# Heating Piping & Air Conditioning

DESIGN
INSTALLATION
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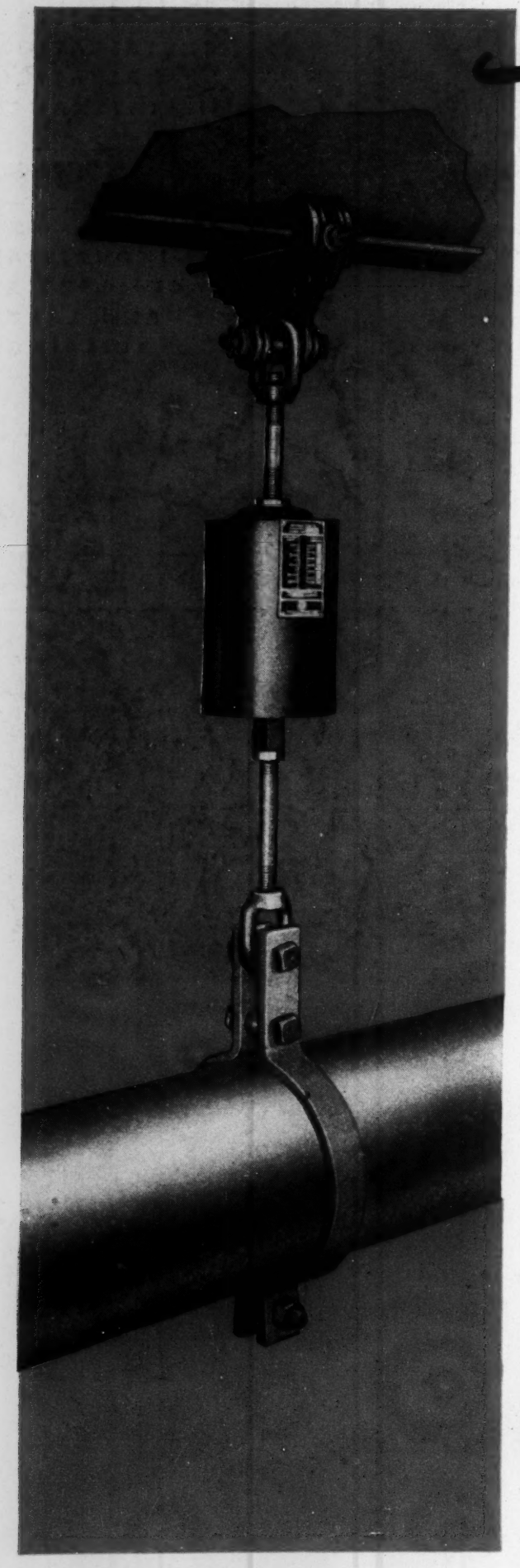
MAY ★ 1944

INDUSTRIAL
COMMERCIAL
AND LARGE
BUILDINGS

297: Journal Section of the American Society of Heating and Ventilating Engineers

*Discussions at the Midwest Power Conference of direct interest to heating, piping, and air conditioning men are given this month*

CONTENTS . . PAGE 69



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# CONTENTS THIS MONTH


May 1944 Vol. 16, No. 5



The Editor's Pages (news notes and comment) ..... front section 65

Corrosion as a Factor in Piping Maintenance, by L. G. Vande Bogart ..... 255

Evaporative Coolers for Diesel Engines, by William Goodman ..... 260

Place of Electronics in Power Generation, by J. D. Ryder ..... 264

Planned Maintenance Systems Keep Industrial Production Up, by William A. Perry ..... 267

Necessity for Research on Combustion Rates and Ash Behavior, by A. R. Mumford ..... 269

Corrosion Inhibitors in Condenser Tube Alloys, by Austin R. Zender and C. L. Bulow ..... 273

Design and Control of Year 'Round Comfort Air Conditioning Systems, by William Goodman ..... 278

Planned Conservation Program Saves Fuel at Large War Plant, by W. S. Bard ..... 283

Measuring the Flow of Fluids (comment) ..... 287

"Open for Discussion" (postwar planning; the metric system) ..... 288

Sam Lewis' Page (central hot water heating) ..... 289

Heating System Control Pays Dividends in War and Peace, by P. A. Hyde 290

Names for Things (comment) ..... 292

Radiant Heating (comment) ..... 292

Foundrymen's Group Approves Code Recommending Ventilation Practice .. 293

WPB . . OPA . . . PAW . . . Etc. (new government orders) ..... 294

Meetings & Conventions (for your calendar) ..... 296

Equipment Developments ..... back section 171

Details of Next Season's Fuel Oil Rationing Program ..... back section 190

New Books & Reports ..... back section 201

Recent Trade Literature ..... back section 205

Index to Advertisers ..... back section 210


## Journal Section of the American Society of Heating and Ventilating Engineers


Semi-Annual Meeting 1944 ..... 297

Some Effects of Attic Fan Operation on Comfort, by W. A. Hinton and W. G. Wanamaker ..... 298

The Resistance to Heat Flow Through Finned Tubing, by W. H. Carrier and S. W. Anderson ..... 304

Summary of Local Chapter Meetings ..... 321

Candidates for Membership ..... 326

Candidates Elected ..... 328

Officers, Council, and Committee on Research ..... 330

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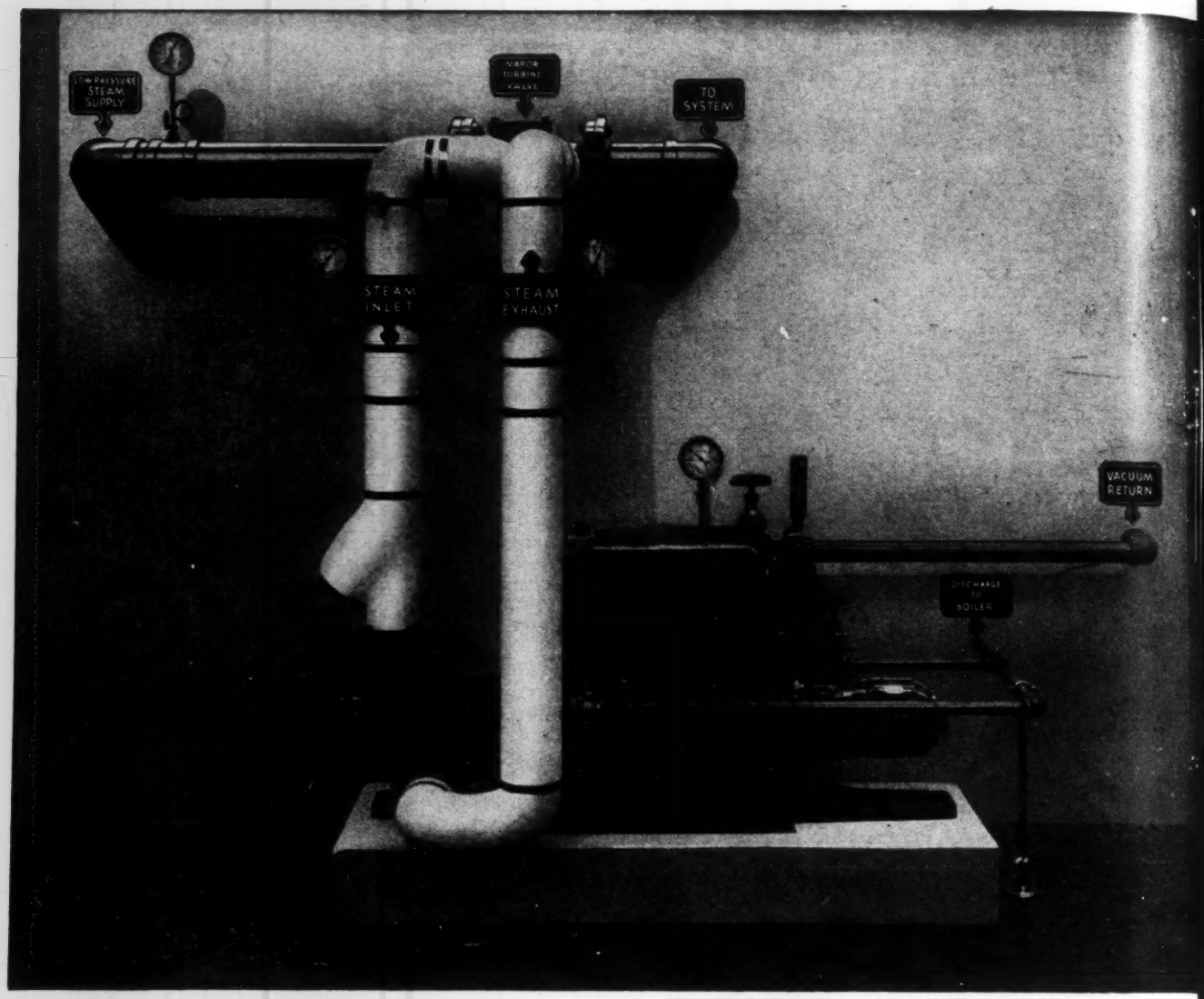


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# Corrosion of Piping and Condenser Tubes . . . Electronic Controls . . . Evaporative Coolers . . . Research in Fuels . . . Planned Maintenance

POWER promotes victory and planning promotes prosperity were the themes of the Midwest Power Conference held at the Palmer House, Chicago, April 13 and 14. With a registration approaching the 800 mark, this year's gathering was one of the largest and most successful since the affair was reorganized in 1938. It was sponsored by the Illinois Institute of Technology with the cooperation of eight other colleges and universities and eight engineering societies. Stanton E. Winston, of Illinois Tech, was the conference director.

The purpose of the conference was to provide an opportunity for all persons interested in power production, transmission, or utilization to meet together for the consideration of mutual problems, with academic sponsorship permitting the freest possible discussion." On the following pages, the papers of particular interest to Heating, Piping & Air Conditioning's readers are given either in whole or in part. Numerous other discussions on subjects not directly within the field of interest of heating, piping, and air conditioning engineers and contractors were also on the conference program.

## Corrosion as a Factor in Piping Maintenance

L. G. VANDE BOGART, research engineer, research and development laboratories, Crane Co., discussed Corrosion as a Factor in Piping Maintenance—With Special Reference to the Physical Effects of Fluid Flow:

Modern piping systems differ from those of a century or more ago in many ways, one of the most important of which is the increase in the number and variety of purposes for which they are now being used. The transportation of water —and later steam—was for years the exclusive reason for which pipe lines were built. Today piping systems are designed and constructed to handle an increasingly large number of fluids for a wide variety of purposes, of which the problem of transportation is just one. Furthermore, they are expected to do this throughout a range of pressures and temperatures limited only by the physical properties of the available materials, and in spite of the fact that

MIDWEST
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many of the fluids are actively corrosive to the more widely used materials.

As a phase of the handling of fluids, engineers have become familiar with their physical properties and with the laws which govern their flow through pipe lines. To many of these same engineers, corrosion is a mysterious phenomenon, the treatment of which they are inclined to leave to metallurgists and chemists or—not infrequently—to the plant maintenance crew. One of the objectives of this paper is to direct the attention of engineers to the fact that—as far as piping maintenance is concerned—the practical solution of many corrosion problems does not depend entirely on the study of electrochemistry and the chemistry of piping materials. In many situations the physical properties of the fluids themselves and the physical effects of fluid flow exert a controlling influence on the course of corrosion, and other factors which are as closely allied to engineering as to chemistry may also be involved.

From the standpoint of a power conference, the corrosion problems encountered in the operation and maintenance of piping systems can be divided into two broad groups. These are: (1) those encountered in central station power plants; and (2) those encountered in industrial plants.

The corrosion problems in the central station plant are almost exclusively those associated with the handling of water and steam, and the successful operation of these plants depends on keeping corrosion as a *potential* rather than an actual cause of piping maintenance. Furthermore, this must be done under circumstances which restrict the methods by which it can be accomplished. Operating temperatures and pressures, and the resulting physical requirements, eliminate some materials which might be acceptable from the standpoint of their ability to resist corrosion. Economic considerations eliminate other materials. In general, therefore, the emphasis in these plants is on corrosion prevention by control of the environment. Careful and thorough water treatment is accepted practice, and the operators of these plants are well acquainted with the necessity for and methods of applying water treatment procedures.

In industrial plants—and to these might be added installations used for non-manufacturing purposes—a different situation exists. Water treatment may follow the pattern set in the central station plant, but it does not always do so. Specifically, it may not—and frequently does not—include *complete* provision for the prevention of corrosion. In these plants, therefore—in contrast with the central station plant—corrosion by water and steam frequently becomes an *actual* and important cause of piping maintenance. In addition, these plants may have as many process corrosion problems as there are processing operations and process fluids in any plant. Treatment of these environments to prevent corrosion is not unknown; but usually it is impracticable and sometimes it is impossible. Here, on the other hand, a considerable latitude in the choice of piping materials is permissible; and the maintenance crews in these plants are therefore forced to contend with a wide variety of metal-fluid combinations.

### Apparent Failure of Materials to Behave as Expected

Detailed information regarding both materials and fluids can be found in the corrosion literature, and if these data are properly used by designing engineers it would seem that corrosion failures in service and the resulting need for maintenance could be avoided. As maintenance engineers well know, this is an optimistic viewpoint. The fact that corrosion does occur, however, does not necessarily mean that the available data have been overlooked or misapplied. In many instances—particularly in piping systems—there is considerable justification for the opinion that corrosion failures are due not so much to the failure to properly apply the available data as to the apparent failure of the materials or fluids to behave in the manner expected of them. The following discussion will call attention to some of the reasons why these deviations from the normal do occur.

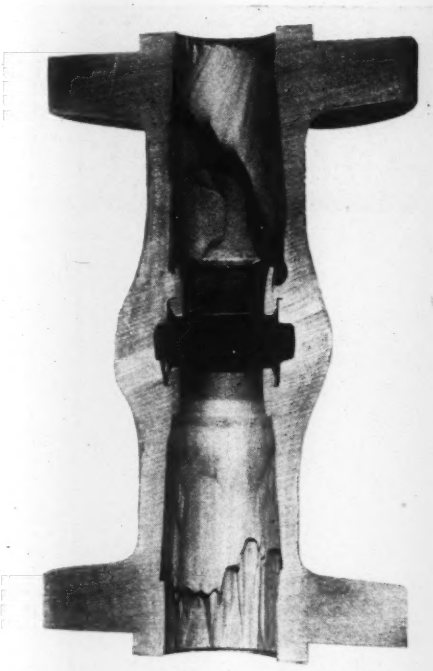
An important feature of the corrosion of metals—at least as far as the common piping materials are concerned—is the fact that the inherent tendency of a metal to corrode is not necessarily a measure of the extent to which it will corrode in service. If this were not true, many of the present day applications of piping materials would not be possible. To a considerable extent the usefulness of these materials depends on the fact that—in contact with the fluids with which they are used—they corrode initially and thereby acquire coatings of corrosion products which are capable of retarding or stopping further corrosion.

**Section of disc removed from a cast iron gate valve used in cooling water line to a condenser. Bronze disc ring corroded under a layer of black iron oxide carried to that point. Adjacent cast iron was not seriously corroded**

From the standpoint of the development and behavior of these coatings there is a close analogy between corrosion and the flow of

fluids. Studies of the latter have demonstrated that the flow of any fluid through a pipe can take two forms, and that there is always a boundary layer of fluid adjacent to the pipe wall which is moving in viscous flow even though the rest of the fluid within the pipe is moving in turbulent flow. Similarly, studies of corrosion—at least by aqueous solutions—show that there is a thin layer of solution adjacent to the corroding surface which differs in composition from the remaining solution. In the specific case of an aqueous solution flowing through a pipe, this layer appears to coincide with the boundary layer just mentioned; and again it can be said that if this were not so, most of the common piping materials would be considerably less useful than engineers have found them to be.

These liquid films are not an exclusive characteristic of the corrosion of piping, however. Theoretically they should exist at any metal surface in contact with an aqueous solution which is capable of corroding that surface, and they are important for two reasons. In the first place, they provide the zone in which occur the reactions which result in the formation of the aforementioned protective coatings. Typical of these are the carbonate scales which form on the inside of domestic water pipes, the phosphate scales which form as the result of certain types of boiler water treatment, and the sulfate coatings on which lead and iron depend for their ability to resist corrosion by dilute and concentrated sulfuric acid, respectively. In the second place these liquid films can and frequently do provide an effective buffer layer between the solution and the metal (or protective coating on the metal). In this capacity their effectiveness depends on such physical factors as thickness and viscosity of the film and on such chemical properties as its pH value and general composition. Since these films form as a result of corrosion, their chemical properties depend on the particular metal-solution combination involved; and since the potential number of these combinations is infinitely large, there is no point in attempting a detailed discussion of them in this paper. The important consideration is the fact that these liquid films play an important part in preventing corrosion and that the study of their physical behavior in any given situation may be just as important as the study of the chemical reactions which result in their formation.



**Interior surface of steel gate valve installed in boiler feedwater line between deaerator and another valve which controlled the flow from the boiler feedwater heater. Adjacent piping not seriously corroded. Severe erosion attributed to impingement of turbulent stream of hot water, which stream carried entrained air bubbles**

In the field of high temperature corrosion by gases and nonaqueous liquids, a somewhat similar condition exists. Here a different type of reaction is involved—one which should not be confused with the electrochemical corrosion reactions commonly encountered in the corrosion of metals by aqueous solutions. In contact with hydrogen sulfide and with oxidizing environments (air, superheated steam, etc.) solid corrosion products are formed directly and adhere to the metal surface in the form of scale. Here, again, the reaction products can and do play an important part; for frequently the serviceability of the metal depends largely on the type of scale formed. If this scale is of the cellular or discontinuous type the environment will diffuse through the scale as it forms and attack will be rapid and will proceed at a fairly uniform rate. If the scale is of the compact or continuous type, the rate of diffusion through the scale will be slow and will become progressively slower as the scale becomes thicker. Chromium is outstanding among the metals with respect to its ability to develop dense adherent oxide coatings; and the fact that this same feature is a characteristic of many of its alloys is in large part responsible for the usefulness of these materials in oxidizing environments at high temperatures. These coatings of scale—whether protective or nonprotective — always provide visual evidence that some corrosion is occurring. Unfortunately, they do not form in all cases of high temperature corrosion. This is particularly true in the case of exposure to hot hydrogen and similar reducing gases, and corrosion by these environments, therefore, is more insidious than that by oxidizing ones.

High temperature corrosion by nonaqueous liquids (e. g., oils and other hydrocarbons) follows the pattern set by gases to the extent that these fluids contain sulfur or other scale forming elements. In addition, or in the absence of such constituents, still another type of protective coating is obtained. The "cracking" which occurs when such fluids are heated above some critical temperature is often accompanied by the deposition of layers of carbon on metal surfaces, and these layers may also afford considerable protection against corrosion. Like certain forms of boiler scale, they form as a result of physico-chemical changes within the fluid rather than as a result of corrosion, although corrosion may affect the degree to which they adhere to the metal surface. Like

these boiler scales, also, they may interfere with heat transfer and have other undesirable properties, but they do have the ability to protect against corrosion.

### Physical Effects of Fluid Flow Disturbing Factor

If the ability of a material to resist corrosion by any environment depends on the formation of protective coatings as a result of contact with that environment, it is obvious that any factor which prevents such a coating from functioning in its normal manner can be the controlling factor in failures due to corrosion. Experience indicates that something of this sort is involved in many of the abnormal examples of corrosion; i. e., when a material behaves in the manner expected of it throughout most of an installation but fails unexpectedly at some few locations. In the case of piping systems—which are designed to control the flow of fluids—experience indicates further that the physical effects of fluid flow frequently provide the disturbing factor.

An example of this with which power plant operators are familiar is provided by the impingement corrosion of condenser tubes, where the pitting of the inlet end of the tubes is attributed to the impingement of a stream of water carrying entrained air bubbles. The water is no more corrosive at that point than at any other point along the tube length, nor is the metal at that end any less resistant to corrosion. The characteristic effect is due to the fact that the impinging air-water mixture either disrupts the liquid film or washes away any solid protective coating which might form. Further along in the tube, where the water moves in conventional manner, the customary layer of corrosion product forms on or adjacent to the tube surface and corrosion may be negligible.

The fact that the above result is obtained with a system which involves both a liquid phase and a gas phase is significant. There is considerable evidence that such two phase systems are apt to be much more damaging than a single phase system involving either component by itself and that this is particularly apt to be true at locations of maximum turbulence. These systems may take the form of bubbles of gas entrained in a liquid, as in the example just cited; or they may take the form of drops of liquid entrained in a stream of gas, as when steam contains drops of water in the liquid state. A common example of damage resulting from the latter is the grooving of valve seats which is frequently called "wire drawing." In almost all such cases the abnormal corrosion is not attributable to mechanical erosion of the metal itself, but —as in the case of impingement corrosion of condenser tubes—is the result of the erosion of otherwise protective coatings or films. Among the exceptions are those cases in which the liquid carries abrasive solids in suspension in which case there may be erosion of the metal itself.

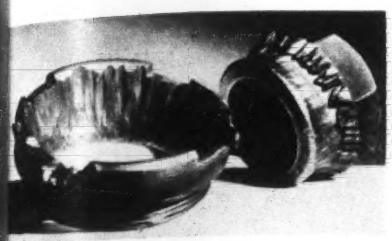
Instances of this type of attack are frequently observed at locations where a phase change is occurring, as in distillation and condensing operations where a liquid changes to vapor, or vice versa. Here the volume changes which accompany the phase changes alter the flow characteristics completely, and the resulting turbulence, combined with the effect of the mixed phases, often increases the corrosion rate far beyond what is observed on either side where the fluid is all liquid or all vapor.

Another type of two phase mixture is that which is encountered in some of the newly developed catalytic processes. The catalysts employed are sometimes acids which are corrosive to many metals at temperatures below the dew point, where aqueous solutions can exist. The flow of fluid through the catalyst mass frequently results in a pickup of dust-like particles of the inert material employed as the catalyst carrier, and this may affect corrosion in two different ways. In the first place these particles of catalyst carrier are distinctly abrasive, and repeated exposure of metal surfaces to vapors carrying these particles and moving at high velocity can result in removal of metal or of any protective coating which might normally tend to prevent corrosion. Experience indicates that this does occur, for some surfaces exposed to such environments have a distinctly sandblasted appearance and are entirely free from scale of any sort, even though the environment and operating temperature might normally be expected to produce scale. In the second place, these solid particles are saturated with catalyst which may be potentially corrosive. When this type of catalyst is employed, when it is carried to and deposited on a metal surface which can acquire no protective coating, and when this surface is alternately cooled down below the dew point of the particular liquid involved, corrosion can be quite general and is sometimes unexpectedly rapid. Both of these manifestations are particularly apt to be encountered at valve seating surfaces—the former when a valve is used to throttle the flow of a fluid containing the entrained solid particles, and the latter when valves are closed while one or both seating surfaces are covered with a layer of the solid material, which is thereby trapped between the two faces.

This experience with catalysts is

### A PROFESSION

**F. M. Dawson, dean of the college of engineering of the State University of Iowa, in his response to the address of welcome at the Midwest Power Conference, made a plea for the consideration of engineering as a profession rather than an occupation, referring particularly to the War Manpower Commission's listing of engineering as an occupation instead of among the professional groups. No action now is indicated, said Dean Dawson, but engineers should plan for public recognition of engineering as a profession in the future.**



**Disc and body seat ring from globe valve used for throttling flow of hot oil carrying suspended sand particles**

typical of a large number of cases of corrosion which might be attributed to failure to properly define the environment. In the example just cited, common practice is to specify material to resist the fluids being processed, most of which are considerably less corrosive to piping materials than are some of the catalysts employed. This is rather typical of what happens in many other cases. As an example, most of the common industrial gases are comparatively inert at temperatures above the dew point (and below some critical temperature at which high temperature corrosion starts) and they are similarly inert at temperatures below the dew point when moisture is absent. Specifications are frequently written on the assumption that moisture is absent or that if it is present, the temperature will remain above the dew point.

The former assumption represents an ideal condition that is seldom attained. The latter may be true as a general condition, but there are few systems in which water does not condense at some location at some time. Associated with this is the further fact that a closed valve often provides an excellent location for entrapment of condensate. When water in the liquid state is present some of these gases become actively corrosive. A rather troublesome feature of corrosion under these circumstances results from the fact that some of the alloys which are outstanding as valve trim materials for use with steam and other gases at elevated temperatures are not so resistant to corrosion by aqueous solutions of the same gases. The straight chromium alloys (12-14 per cent Cr) for example, are extremely susceptible to corrosion by aqueous solutions of sulfur dioxide and/or hydrogen sulfide, the constituents which are so commonly encountered in the gaseous products of combustion of coal and oil and also in natural and petroleum refinery gases. As a consequence, more than one perfectly good steam valve has failed in steam service on soot blowers because of corrosion by sulfur dioxide dissolved in the condensate which collects after the valve is closed. Similarly, many an oil or gas valve has failed because of corrosion by aqueous solutions of hydrogen sulfide, particularly at locations where a gas containing "negligible" quantities of the corroding agent has been compressed to a point where the concentration (on a volume basis) of the same component is objectionably high.

### Air Inleakage and Corrosion By Water

The law of averages would suggest that the failure to properly define an environment might occur most frequently among the most commonly encountered fluids, and again experience indicates that this is true. Obviously, the most common one is water, and in spite of all that has been said and written about corrosion by water and about methods for preventing such corrosion, piping systems carrying water are still corroding. There seems to be a tendency to assume (1) that water from any given source is non-corrosive until proved otherwise and (2) that because water from one source is largely trouble free in one installation it will likewise be trouble free in another installation operated under entirely different conditions. Unfortunately, the reverse is true in many cases. In Chicago, for example, there are industrial plants and large buildings using city water and in which maintenance costs are kept at low figures by the use of rather simple water treatment and even when no treatment is employed. In other buildings—perhaps just across the street or in the next block—corrosion is rampant. In general, corrosion in these buildings is more apt to occur in return lines, although corrosion of boilers is not unknown. To the extent that the remedy for such corrosion lies in water treatment, this paper has no comments to offer, as the whole subject of water treatment has been discussed in considerable detail before previous conferences. While its importance has been emphasized repeatedly, however, there is still considerable evidence that water treatment to prevent corrosion is still being omitted in situations where it might be useful.

There is another phase of this whole situation which does deserve some comment, however. The prevalence of corrosive condensates in buildings and groups of buildings which have been constructed during recent years under emergency conditions suggests something which may be of particular interest to maintenance engineers. Many of these installations are spread over wide areas and supplied by central steam generating plants. Both supply and return lines are longer than in more compact installations; hence the total number of joints is correspondingly larger. In return lines there may be no visual evidence that joints are not tight; yet analysis of the condensate at successive locations may indicate that air inleakage is occurring. The nature and location of corrosion in such lines points—in some cases—to the fact that air inleakage is largely responsible for converting a potentially corrosive condensate into one that is actively so.

The foregoing discussion has called attention to a number of physical and mechanical factors which are involved in the corrosion of piping systems, and there are still others which fall in the same category. Time and space limitations do not permit consideration of the more abstract chemical and electrochemical factors involved. The emphasis, therefore, has been put on those factors which fall

within the scope of the activities of operating and maintenance engineers. The final and successful solution of any corrosion problem usually depends on correlation between both groups of factors. If they are not properly correlated it may appear that "theory is theory and practice is practice—and never the twain shall meet." When the engineers who know their plants and what transpires therein join hands with the chemists and metallurgists who may be better acquainted with the corrosion characteristics of fluids and of materials, then theory and practice can and do meet.

# Evaporative Coolers for Diesel Engines

EVAPORATIVE Coolers for Diesel Engines were discussed by William Goodman, consulting engineer, the Trane Co. (and a member of HPAC's board of consulting and contributing editors):

In a closed system the same water is continuously recirculated between the engine jackets and a coil. The water is heated in the engine jacket and is then cooled by a coil separate from the engine. Once the jacket water surrenders its heat in the cooling coil, there are several methods of carrying away this heat from the coil. For example, raw city water or raw well water can be circulated over the outside of the cooling coil, or the coil can be placed in the bottom of an atmospheric cooling tower with the spray water dripping over it. Finally, the coil can be placed in the casing of an evaporative cooler.

All of these methods of cooling the coil have one thing in common, and that is that the raw cooling water never comes in contact with the relatively soft water circulated between the engine jacket and the inside of the cooling coil. Because of this, the same water is continuously recirculated through the engine jackets month after month. In this manner, all possibility of depositing scale in the water jackets of the engine is avoided. Even if a relatively small amount of raw make up water is added continually as is done in an open system, the amount of scale that can be deposited in the water jackets in a short time is amazing. So great is the danger to engines from the scale deposited by even the small amount of make up water required for a cooling tower that insurance rates for open systems in which continuous make up is required are higher than in the much safer closed systems requiring practically no make up water.

In a closed water circulating system using an evaporative cooler, the cooling coil is placed in a vertical casing and raw water is sprayed from above over the outside of the coil. The water drips through the coil from top to bottom completely wetting all of the surfaces. In addition, air which ultimately carries off the heat from the soft water inside the coil is drawn over the outside of the coil. Generally the air is drawn from the bottom of the coil up through the top counterflow to the water. Picture then a coil with soft water flowing through the inside of it. A rain of raw water is falling over the outside from top to bottom, while against this rain a stream of high velocity air is drawn up through the coil. There you have in essence an evaporative water cooler.

As the soft jacket water flows through the inside of the cooling coil, its temperature falls. On the other hand, the temperature of the raw spray water on the outside of the coil barely changes at all. The temperature of the spray water from the top to the bottom of the coil is always practically constant. The main function of the spray water is to keep the outside surfaces of the coil wet so that the heat coming through to the outside of the coil can be carried off by evaporating some of the spray water. Hence the same spray water with a small amount of make up can be recirculated continuously. It is the evaporating spray water that carries off the heat and not the portion of the spray water that falls to the water pan at the bottom of the cooler. The air is provided only as a means for rapidly removing the evaporated moisture. Although the air temperature rises, the major part of the heat is carried off by the evaporated moisture. Primarily, the air does not carry off the heat from the jacket water by an increase in the temperature of the air. Instead, the main function of the air is to remove rapidly the moisture evaporated from the outside surface of the coils.

### Gpm and Hp Requirements

You may wonder why I so carefully emphasize the fact that the heat removed from the soft circulating water is carried off by evaporating some of the spray water. The importance of this statement lies in the fact that the quantity of water that must be sprayed over the outside of the coil is not determined by any considerations involving its temperature rise. As you are probably aware, in an atmospheric tower with a cooling coil in its base, the water sprayed over the top of the tower must actually fall in temperature before it reaches the cooling coil in the base. Then when the spray water reaches the coil in the base, the temperature of the spray water is increased as it absorbs the heat from the warm jacket water inside the coil. The warm spray water is then picked up by a pump and sprayed over the top of the tower to be cooled again.

As an example, such a tower with a coil in its base for a

MIDWEST
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1000 hp engine would require about 600 gpm of spray water if the rise in the temperature of the spray water is to be limited to 10 F as it flows over the coil in the base. For an evaporative cooler, on the other hand, the amount of spray water for such an engine would be only about 100 gpm. The reason for this is that heat is not removed by a rise in the temperature of the spray water. The function of the spray water is, as I have said before, only to keep the surfaces wet, and for this purpose 100 gpm is several times as much as is actually needed. However, the larger quantity is provided because its vigorous flushing action helps to keep the surfaces of the coil clean and free from dirt.

### WAR TIME

**It has been variously estimated that the peak electrical load has been reduced 800,000 to 1,500,000 kw by the adoption of war time in the United States, according to Alex D. Bailey, assistant to the vice-president of Commonwealth Edison Co., in his talk on postwar planning of the nation's power supply at the Midwest Power Conference. The winter peak in the Chicago area has been reduced at least 80,000 kw, he said, due to war time.**

You may realize most clearly the small quantities of spray water that are required for an evaporative cooler when I tell you that the pump for circulating the spray water in an evaporative cooler for a 1000 hp engine requires only a 1 hp motor. The pump for an atmospheric tower would under the same circumstances require a motor of about 7½ hp.

Of course, in both the atmospheric cooling tower and in the evaporative cooler, all of the heat is ultimately removed by evaporation of a certain amount of moisture but in an atmospheric tower, there is an intermediate step. The spray water must first be cooled by evaporation and then the heat removed from the soft water by an actual rise in the temperature of the spray water. In the evaporative cooler, we skip the intermediate step in which the temperature of the spray water rises. It is for this reason that a very much smaller quantity of water needs to be circulated in an evaporative cooler than in an atmospheric tower with a coil in the base or in an atmospheric tower with a shell and tube cooler.

Just the quantity of water that must be sprayed over the cooling coil is considerably less in an evaporative cooler than in any other type of cooling system. The quantity of water that must be circulated between the engine and the coil is, of course, the same in all types of cooling systems. The quantity that is to be circulated in distinction to the quantity that must be sprayed is determined altogether by the heat to be removed from the engine and the allowable temperature rise of the water. The amount of heat rejected by the engine depends on the purpose for which the water is used. If the water is used only to cool the cylinders, then it must pick up about 2500 Btu per bhp. However, if the water is also used for cooling oil, then approximately an additional 800 Btu per bhp must be removed, making a total of about 3300 Btu per bhp. Finally, if the exhaust is water jacketed, an additional 200 Btu per bhp must be removed. All in all, depending upon the make of engine, the amount of heat that must be removed per bhp ranges from about 2500 up to 3800 Btu per bhp. Although 3000 Btu per bhp is used for rough preliminary estimates, the range in heat that must be picked up by the water is so great that before selecting a cooler the amount of heat that is rejected to the water should be obtained from the builder of the engine.

Although the quantity of heat to be absorbed by the cooling water should be obtained from the engine builder, the temperature rise of the water can be fixed by the purchaser within reasonable limits. Temperature ranges of from 10 F up to as much as 30 F have been used. However, 30 F is a rather wide range. Most systems have been installed with temperature changes of from 10 to 20 F with preference being given today to the lower temperature rise of 10 F in spite of the greater quantity of water required. The small rise of 10 F is desirable in an engine because better heat transfer and much better circulation is obtained in the water jackets with the larger quantity of

water needed to hold the temperature rise down to about 10 F. In addition, the larger quantity of water lessens the possibility of a stagnant pocket in the jackets where the water circulation is poor. In such stagnant pockets high local temperatures may be encountered. In fact, steam may even be formed in various spots when poor circulation is obtained. Finally, the smaller temperature spread of the water means that the entire engine will be at a much more uniform temperature, thus eliminating stresses due to widely differing temperatures.

Whether the temperature rise of the water should be 10 or 20 F depends also to some extent upon the power required to circulate the water. For a 1000 hp engine, only 300 gpm would have to be circulated for a 20 F rise, whereas 600 gpm would be required for a 10 F rise. There is considerable gain in the effectiveness of heat transfer as the water quantity is increased from a small amount to a larger amount. However, once a respectable amount of water has been obtained, increasing the amount of water further does not very much increase the effectiveness of heat transfer, but the pressure drop and the power required to circulate the water rise very rapidly. Hence, before arbitrarily selecting a 10 F rise, it is well to check the power that is required to circulate the water. Wherever possible, preference should be given to the quantity of water required for only a 10 F rise, but sometimes the difference in power required between, say, a 15 F rise and a 10 F rise may not warrant providing enough water for only a 10 F rise. In some installations the increased heat transfer with the quantity of

MIDWEST
# POWER CONFERENCE

water required for a 10 F rise may be only very little better than with the quantity of water required with a 12 F or 15 F temperature rise, whereas there may be a considerable saving in power with the larger temperature rise. The power required is, of course, only one of the factors to be considered, but it should not be ignored.

### Temperature of Circulating Water

It is well here to emphasize that for a given engine only the temperature rise of the water determines the quantity of water that must be circulated. The actual temperature of the water has no effect upon the quantity of circulating water needed. Thus for a 10 F rise the same quantity of water must be circulated between the engine and the cooler regardless whether the water leaves the engine at a temperature of 120 or 160. However, the temperature of the circulating water does have a very marked effect upon the size of the evaporative cooler that is needed for a given engine. The higher the temperature of the engine water, the smaller will be the evaporative cooler with a corresponding decrease in investment and operating costs. Thus for a 1000 hp engine, a bare evaporative cooler for a 10 F water rise with jacket water off the engine at 120 F will cost approximately $3300. If the jacket water temperature could be 150 F, the cost of the bare evaporative cooler would be reduced to about $2300, and for 160 F to about $1900. The operating cost would also be slightly smaller because for 120 F water the total power required for the fan and spray water pump would be 8½ hp, whereas for 150 F water the total horsepower would be 6 hp. In addition, the unit for 150 F water would require slightly less space.

I realize that the great majority of engines in service today operate with water temperatures off the engine of less than 120 F. This temperature of 120 F seems to be a survival of the early days when most engines were cooled by raw water being circulated through the jackets. Under these conditions, it was necessary to keep the temperature of the water off the engine as low as reasonably possible in order to minimize the deposit of scale. The lower the temperature of the jacket water, the more slowly the scale would deposit. However, with a modern closed system, it does not appear necessary to hold the water temperature down to 120 F. Indeed, it has been reported that engines have been operated successfully and without any lubrication difficulties with vapor-phase cooling in which jacket water temperatures exceed 200 F. Boyer states that during the last few years several large engines have been operated with water temperatures off the jackets of 175 and 180 F. He also states that too low water temperatures can be just as injurious to an engine as excessive temperatures.

I am not an expert on the design and operation of diesel engines and, therefore, cannot, of course, pretend to recommend jacket water temperatures to you gentlemen. The proper jacket water temperature should be set by the manufacturer of the engine. However, the general opinion in the industry today seems to be that the higher jacket water temperatures are beneficial to the operation of the engine. Indeed, Arthur Judge in his book on High Speed Diesel Engines presents a chart showing test results on such an engine. As the temperature of the jacket water off the engine was increased from 120 to 180 F the brake horsepower of the engine increased from 91 to 94, an increase of 3½ per cent. But the effect on fuel consumption was even more marked. The pounds of fuel per brake horsepower per hour required at 180 F was 7 per cent less than at 120 F. Hence, from all the evidence available, it appears that higher jacket water temperatures are definitely beneficial to the operation of the engine. Insofar as the evaporative cooler is concerned, the use of higher jacket water temperatures would mean a lower investment in the cooler, lower operating costs, and smaller space requirements.

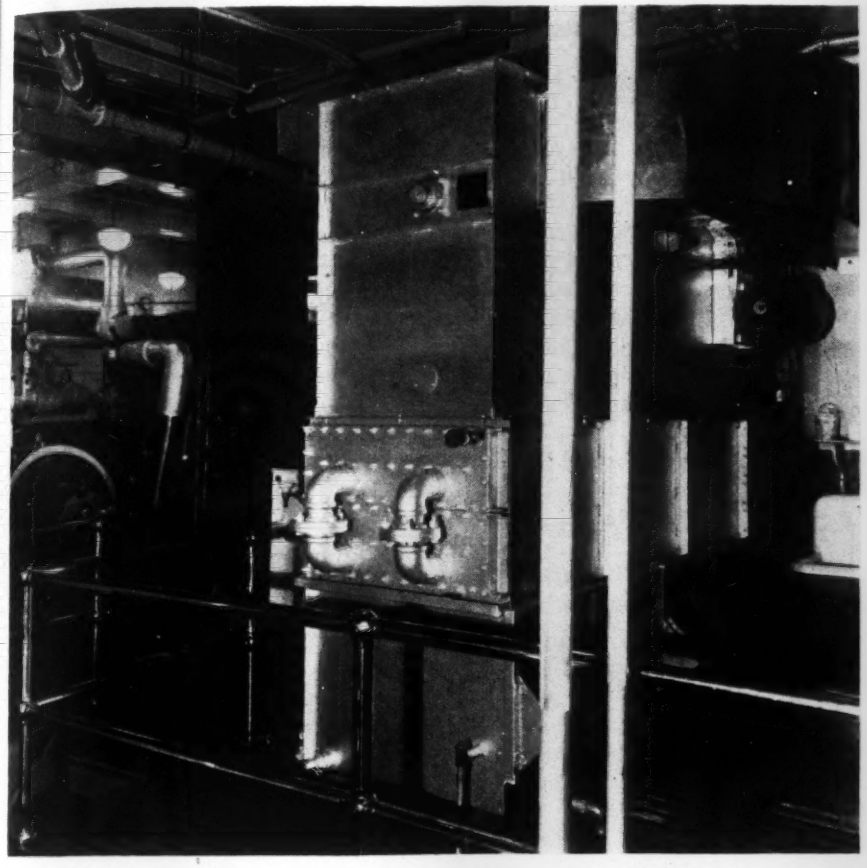
### Control of Jacket Water Temperature

For good operation, control of the temperature of the jacket water is desirable so that the engine is always operated at a nearly uniform temperature. Although the control of water jacket temperature is relatively simple, it is somewhat complicated in cold climates by the necessity of avoiding a freeze-up of the spray water. In climates where freezing temperatures are not encountered, there is, of course, no problem. A three way valve bypassing part of the water around the evaporative cooler is all that is needed. Another method is the use of a thermostat to control throttling dampers which reduce the flow of air through the evaporative cooler. This method, while effective, is generally not as satisfactory under light loads as a three way valve. This is especially true of an evaporative cooler using forward curve fans which, when the air volume is reduced greatly, will cause some surging of the air. This, however, is not serious under ordinary conditions of operation.

When a three way valve is used, it should be located on the outlet pipe from the cooling coil so that two streams of water flow into the valve and the mixed stream of water flows out of the valve. If an ordinary three way valve is located in such a manner that one stream flows in and two streams flow out, there may be difficulty with the valve snapping to the seat when it is near either extreme position. For this reason, I would like to repeat, when using three way valves to bypass part of the water around the evaporative cooler, locate the valve in such a manner that two streams of water flow into the valve and only one stream flows out.

In climates where freezing temperatures occur, a slightly more

elaborate control system is needed for controlling the temperature of the jacket water. The full volume of air is always drawn by the fans through the evaporative cooler. However, an air bypass permits part of the warm air discharged from the fans to mix with a small amount of outdoor air, and this warmer mixture is then drawn into the intake of the evaporative cooler. In this manner the temperature of the air entering the evaporative cooler is always kept above the freezing temperature. Accurate control is obtained by this method without any danger of a freeze-up.



**Evaporative cooler serving the diesel engine in a large New York skyscraper**

I should have mentioned earlier that the evaporative cooler is usually located inside of the engine room. This greatly simplifies the problem of maintenance, and keeps the entire cooling system under the eye of the engineer where anything unusual can be detected immediately. I need not tell you gentlemen who operate plants of the many advantages of having the evaporative cooler inside of the engine room where it can be kept at a fairly uniform temperature all the year around and where routine maintenance can be performed conveniently.

### Ventilating and Heating the Engine Room

Because of the usual location of the evaporative cooler inside the engine room, it provides an ideal means of ventilating the engine room. By opening windows in the wall at the opposite side of the engine room, a continuous stream of outside air can be drawn through the engine room at all times. When installing an evaporative condenser, always make provision for auxiliary dampers so that ventilation can be provided.

Sometimes an attempt is made to heat the engine room with the warm, moist air discharged from the evaporative cooler. However, this is not desirable because of difficulties with frosted windows. The air discharged from an evaporative cooler is not only warm but is also extremely moist. This usually leads to excessive humidity in the room with excessive condensation on the windows and, in extreme cases, on the walls. For this reason the moist air discharged from the evaporative cooler is not satisfactory for heating. If you want to heat an engine room with waste heat, the most satisfactory method is by means of unit heaters with the warm jacket water circulated through the unit heaters before it goes to the evaporative cooler.

Rarely is the coil in an evaporative cooler large enough to permit the spray water to be shut off completely when the temperature of the outside air falls to 35 F. With the spray water shut off, all of the cooling would have to be accomplished by transferring the heat to the cold air and raising its temperature. Usually, however, the coils in evaporative coolers are not large enough to permit shutting off the water at about 35 F. It is surprising how much the spray water adds to the heat transfer between the coil and the air. A sprayed coil that can easily carry the load with 80 F air may not be able to carry the load with 35 F air when the spray water is turned off. However, where a coil is purchased large enough to permit the spray water to be turned off with air temperatures of 35 F, the air discharged from the evaporative cooler can be used to heat the engine room in subfreezing weather. Furthermore, if the spray water can be turned off when the outside air drops below 35 F, control of the temperature of the spray water can be accomplished by the same simple methods that are used in climates where the temperature of the outside air never falls to 35 F, that is, by means of either a three way valve bypassing water around the evaporative cooler, or by throttling the air.

When installing an evaporative

cooler, no special foundations are needed. The cooler itself is not heavy. For example, for a 500 hp engine, the cooler weighs about 5000 lb, and for a 1000 hp engine about 10,000 lb. This weight is spread practically uniformly over the entire base of the cooler. Consequently, the load per square foot is low. For this reason evaporative coolers can be set on ordinary floors of even 4 in. thick concrete.

Wherever possible, locate the evaporative cooler against an outside wall and as close to the engine as possible. This will eliminate excessive runs of duct work and will keep the runs of piping between the cooler and the engine short, all of which contributes to low installation costs and low operating costs.

### Many Combinations of Surface and Air Quantity

When designing evaporative coolers, it is possible to use many different combinations of cooling surface and air quantity. The greater the amount of cooling surface provided, the smaller the quantity of air that must be circulated. Conversely, the smaller the surface provided, the greater the air quantity that is needed. In other words, there is a choice between air quantity and surface. Most manufacturers have standardized their designs. However, it is possible to obtain units with larger coils and smaller air quantities. You will understand the advantage of such units with larger coils and smaller air quantities in a moment. If, after the installation is completed, it is found that a little greater cooling capacity would be desirable for the evaporative cooler, such additional capacity can easily be obtained from a unit designed initially with a larger surface and a smaller air quantity. By increasing the air quantity of a unit with a larger surface, the capacity can be markedly increased. However, in units which have a small amount of surface and a large quantity of air to begin with, an increase in the already large quantity of air will have little effect upon the capacity of the unit.

Probably many of you have had the experience of trying in extreme weather to obtain additional capacity from the evaporative coolers by speeding up the fans, only to find that very little additional heat dissipation was obtained while the power required to operate the fan increased by a surprisingly large amount. However, in units which start off with an initial small air quantity, it is possible to obtain quite a marked increase in capacity with an increase in air volume. This reserve capacity under some conditions is worthwhile buying.

To replace the evaporated spray water, a small amount of make up water is, of course, continually needed. It is a good plan to provide more water than is actually needed to replace the amount evaporated, thus allowing a small amount of water to overflow to the drain. In this way the spray water is continuously diluted and the concentration of salts due to the make up water can be kept at a fairly low point. The expense is not great because the amount wasted to the overflow need not be very large. The actual amount, of course, depends upon the salt concentration of the raw water used for make up and is best determined by chemical analysis or by experience, as is done in most cases.

### Cooling the Oil

Frequently the oil is cooled by means of a shell and tube cooler. However, it is possible to cool the oil right in the evaporative cooler. All that is required is a separate coil in the evaporative cooler for cooling the oil. The coil for cooling the oil is, of course, considerably smaller than the coil for the water and can usually be added for less expense than a separate shell and tube cooler can be bought. Where the evaporative cooler is located close to the engine, the cost of oil piping between the engine and evaporative cooler will not be a factor.

In this short talk I have tried only to give some of the basic facts about evaporative coolers, how they work, the factors influencing their selection, and something about the details governing their installation. It is possible in a short talk like this to touch only upon the highlights, but I do hope that in some measure I have contributed something to your knowledge about these versatile and useful coolers.

## Place of Electronics in Power Generation

THE Place of Electronics in Power Generation was the subject of J. D. Ryder, assistant professor of electrical engineering, Iowa State College. Following are excerpts from his paper:

Before attempting to find a place for electronics in power generation, it seems desirable that the term electronics be defined and discussed, to place it on a solid engineering basis. Men in other fields may then be in a better position to assay the future effect of electronics on their own field of interest without the necessity of becoming electronic engineers.

There are two ways in which electronics differs from power electrical engineering in its viewpoint of the electron. First, in electronics we are concerned primarily with the actions of free or individual electrons relatively unaffected by their surroundings, while in power electrical engineering we think of these electrons en masse, obeying laws which are statistical averages of the behavior of the separate electrons. These large groups of electrons are affected by conditions in the conductors in which they flow, such as electrical resistance.

A second and possibly more im-

portant difference in viewpoint is with respect to the velocities at which the electrons travel. In electronics we are interested in devices in which electrons travel unimpeded at speeds of thousands of miles per second; in power electrical engineering the electrons rarely move at rates greater than a few inches per second.

Because of the close relationship of the individual electron to electronics we may well ask—what is an electron? Present day knowledge on that point does not go far. We can predict what an electron will do under certain circumstances but know nothing of its shape, size or internal construction. Combustion engineers would be in much the same situation if they knew only the Btu value of their boilers, but would be at a loss to understand what went on in the combustion process.

We do know that an electron is the smallest particle of matter and that all matter contains electrons. We know that it takes $5 \times 10^{29}$ electrons to weigh a pound, an inconceivable number. Our most important knowledge is that an electron carries an electric charge, is in fact our smallest indivisible unit of electricity. This charge is so small, however, that it takes $6 \times 10^{18}$ electrons passing through every second to heat a 100 watt electric lamp to incandescence.

Since an electron is an electric charge, then a flow or movement of electrons constitutes an electric current, whether it be in an electron tube, a lamp filament, or an electric motor.

We have seen that an electric current is the same regardless of the material through which it flows, but it is the difference in the "material" of the path that furnishes us with a simple definition of electronics.

Electronics may be defined as the field of devices which function by reason of passage of electrons through gas or vacuum.

This definition establishes the fundamental distinction between power electrical engineering and electronics as a difference in the nature of the electrical conducting path. It is seen to exclude conduction of electric current through solids or liquids, such as our power transmission and electrolytic processes, but to include all other cases of conduction of electric current (or electrons) such as our conventional radio tubes, gas filled tubes, x ray tubes, fluorescent lamps, neon lamps; and switches and circuit breakers in the power field.

While the latter are not usually thought of as electronic devices, yet their operation is dependent on controlling the flow of electrons through air and they appear to be correctly included within the field of the definition.

### Why Electronics?

It can probably be agreed that an electronic device can be designed to perform the most complicated operation dreamed of by the most highly gifted press agent. An electronic control could be built to operate the most cantankerous boiler, under the most vicious load cycle, holding the steam temperature, with never a waver in the steam pressure line, and having the entire performance of the boiler operator limited to pushing the start button.

Since there are already on the market automatic control systems which approach this ideal, why would the electronic system be preferred? An examination on an engineering basis discloses only three reasons why engineers, as business men, will purchase electronic control or metering devices. These three fundamental reasons are:

1) The electronic system can perform an operation *no other device can do.*

2) The electronic system can perform an operation *better* than any other system.

3) The electronic system is *cheaper* than any other system.

Electronic equipment can often perform operations impossible by any other apparatus. Such equipment (smoke recorders are an example) will find a ready sale at any price consistent with the value of the result, will be bought not because they are electronic but because there is no other way.

Electronic equipment can often perform operations better than some other types of equipment; the new fast, continuous temperature recorders are examples, and will be purchased because a better result is obtained at a price consistent with performance.

Electronic equipment unfortunately is not always cheaper than competing systems. When it is cheaper it will be purchased for obvious reasons.

If a piece of electronic equipment cannot be justified by any of the above, it will not be sold. Newness or novelty does not determine the design of modern boiler and turbine rooms.

To state the matter in another way, electronics can do any job—but it will not always be justified by the engineering and economic factors involved.

Fortunately there are certain factors in the equipment design, which, if understood by the customer and applied at the time of a purchase, may prevent a misapplication and purchase of unsatisfactory electronic equipment.

### Electron Tube Stability

An electrical engineer thinks of a resistor, an inductance, or a capacity as a circuit constant which can be purchased as having a certain value and which will have that same value at all times. An electron tube is not such a precision device. When new, they are considered satisfactory if they do not vary more than about plus or minus 20 per cent of the standard rated values. During life the tube characteristics are continually chang-

## NAVY NEEDS ENGINEERS

**A Navy lieutenant told the opening session at the Midwest Power Conference that the Navy is still in great need of officers and is still commissioning men with the proper qualifications from civilian life. A college degree and business experience is usually essential, but in some cases two years of college work with the appropriate experience is an acceptable background.**

# MIDWEST POWER CONFERENCE

ing, causing changes in circuit performance, unless the design was given sufficient safety factor. Failure to allow for the electron tube as a circuit variable is one reason electrical engineers with power training frequently have trouble designing and applying electronic devices.

A device may work perfectly with a certain set of tubes at the factory, and later when the customer must replace some of the tubes it becomes inoperative. Another difficulty occurs in not allowing for the change of tube characteristics during life of the tubes, and as a result the calibration of the device is found to be in error later, even though perfectly correct at installation.

It is almost always possible to design electronic devices to be independent of the characteristics of the tubes used. As a rule it calls for a higher order of engineering ability and may raise the cost slightly. The results will be found worth the additional cost, since a well engineered design is indicated.

A design of this nature means that a tube may change its characteristics over a very considerable range without affecting the performance. In one example a tube operated for over five years, or about 45,000 hr. When removed the tube tested extremely low, far below the point at which it would be usable in a radio receiver, yet at no time had the recorder shown any calibration inaccuracies. When a new tube was inserted, the recorder continued to operate without change.

Performance substantially independent of tube characteristics is especially important in a control. A change in tube characteristics may cause a shift in the control point, or a tube failure may cause the control to run away if the design is not made independent of the tubes.

If the operating instructions mention periodic checking of calibration or provide an adjustment to be checked periodically, then the instrument is probably not designed to be independent of the tubes. As a check, insert various tubes and note the effect on the calibration, and for a control, remove one of the tubes, simulating a complete tube failure, and note the effect on the control operation. Then ask, is that effect desirable?

The weak spot in many electronic applications is not in the electronic circuits but in the mechanical parts used to actuate the electronic control, or in the mechanical parts used to perform the desired operation. These are the features that the electronic engineer will know least about, because they may be peculiar to one application.

In an application of electron tubes to a printing register control it took six weeks' work, and a complete redesign of part of the machine to make the electronic register control function. The customer's application engineer failed to allow for the instantaneous, high speed action of the control and had to reduce weights and strengthen parts before the mechanical part of the machine would stand up. During this time the electronic circuits operated perfectly at all times.

### Possible Applications

Having surveyed some of the factors which should underlie the application of electronics to any field, let us now ask—what is its future in power generation? If we remember the three basic reasons for the application of an electronic device we can find a number of present and future applications in which electronics will definitely be a factor.

*Smoke*—Smoke measurement and recording happens to be an excellent example of an application which can be done in almost no other way than by electronics. There are at the present time several photoelectric smoke recorders on the market, operating on various principles, but essentially they cast a beam of light across the stack, striking a photoelectric cell, and the output current or voltage of this cell is recorded as a function of smoke. When we try to decide what function of smoke this is we get into trouble.

First, what is smoke? Until this question is settled by engineers, independently of the politicians, we will have no really satisfactory measuring of smoke density. Certainly the Ringelmann chart, with its reliance on purely subjective readings by the human eye, affected by the color of the smoke, the cloudiness of the sky, the position of the sun, is not an engineering measurement.

Second, many smoke recorder designers and smoke inspectors seem never to have heard of Lambert's law, which states that the amount of light absorbed by the smoke is an exponential function of the thickness of the smoke column. The smoke from a 20 ft stack will absorb four times the light absorbed by a column of the same smoke in a 10 ft stack, giving a reading of much more smoke when actually there are exactly the same number of particles per cubic foot.

These are examples of some of the difficulties besetting the application of an electronic device in a new field. They are not the fault of electronics, but you will frequently hear electronic smoke recorders blamed for not reading correct smoke densities. What is smoke?

*Temperature*—Another point at which electronics is entering the power field is in the measurement of temperature. For many years the intermittent, step by step type of temperature recorder was standard and did an excellent job. Electronics entered the picture and it was found possible to improve on even the best of the preceding types by elimination of most of the mechanical moving parts, by the elimination of the delicate gal-

### CONSERVE FUEL

**The real need for conservation of all fuels in every possible way was brought out in the course of several of the discussions at the Midwest Power Conference. Coal, oil, and gas are vital war materials, and eliminating waste is a real contribution to victory.**

vanometer, or by removal of the step by step action. There are now recorders on the market retaining all the accuracy of the older models but giving continuous movement of recording pens, without a galvanometer, and with no mechanical parts more complicated or troublesome than an electric motor. Electronics has made possible these improvements, and there are enough potential developments now available to insure a continued advance in this field after the war.

These new developments will take the line of improved versions of temperature recorders using electronic tubes as actuating elements. We can also look forward to the complete end of measurement of radiant temperatures by putting something in the furnace or pipe. The new way will involve electronic means and we will measure temperature by looking at the hot body. We will use such devices not because they are electronic but because they are better.

*Flow*—The use of electronics in the measurement of flow is not so far advanced. Means have been proposed for this but so far they do not look too promising. This excludes the transmission or telemetering of flow readings electronically. The latter field is already well developed, there being a number of devices on the market. We can expect a continued development of electronic transmission devices, especially with the large war use of position indicators, but so far no one has proposed a practical method of eliminating the orifice in the pipe line and of performing the flow measurement electronically and external to the pipe.

The position indicator or flow transmitter is a device well able to make maximum use of the advantages offered by electronics. Electronic devices require very small or zero input torque, they can operate at high speed, and the currents and voltages encountered are suitable for transmission over telephone wires as is frequently desirable. This is an excellent and almost always dependable application of electronics. It would seem that a thorough investigation of the possibilities of such devices should be made by the power industry, especially as to their advantages in transmitting readings to the boiler and turbine recorders and control panels, in contrast to running high pressure piping and tubing into these panels. This will become increasingly important as operating pressures and temperatures rise, both from a safety and a cost standpoint.

*Pressure*—What has been said of flow measurement applies almost equally well to pressure measurement. While we have very satisfactory recorders of dynamic pressures as encountered in reciprocating engines, we do not as yet have a truly electronic method of measuring static pressures as encountered in steam generating stations. There are several possible paths of approach to the problem and it seems reasonable to expect ultimate success, with the line of development taking advantage of electronic means.

*Control*—The field of automatic control offers possibly the largest opening for electronic equipment in power generation. The advantages of zero input torque, high speed, limitless output torque, and almost unbelievable precision should give electronics a great opportunity. It is here, however, that the greatest engineering skill in application will be required if electronics is not to receive an irreparable setback at the very beginning.

The war is giving us electronic control devices in considerable variety, but their application to industrial control will not be as easy as the press agents would have us believe. The requirements of performance, safety, and dependability are very high and it will be in the application, not in the electronic circuits themselves, that the difficulties will be encountered. Watch for them there and allow the electronic engineer to worry about the tubes.

The first applications will probably be made on control problems where speed is an essential. This is an advantage of the electronic system over competing systems which will be capitalized. Another advantage is simplicity of mechanical moving parts, reducing maintenance. The use of electric wires instead of oil or air tubing for transmission will be cheaper and more advantageous in certain cases. But in every case the selection of electronic control will be based on the three fundamental reasons and not because it is electronic.

At the present time several companies can offer partial systems of automatic electronic control, from simple on-off systems to the more complicated modulating forms of control. As the war ends we will find an increasing variety of this apparatus available, possibly involving greater use of more complicated control systems such as second and third derivative control, functions which can be easily introduced into electronic control systems.

## Planned Maintenance Systems Keep Industrial Production Up

WILLIAM A. Perry, superintendent of the electric and power departments of Inland Steel Co., discussed the reasons for planned or preventive maintenance systems, told how they aid in keeping industry going at full production, and explained how such systems are instituted and carried through. Following are excerpts from his paper:

What do we expect to gain or prevent by planned maintenance? How much are we willing to give to put the system into effect and operate under its plans? Why are we all so conscious of maintenance systems at this time? Today undoubtedly the answers to these questions are very likely to be al-

most unanimous; namely, to run 24 hr a day and as many days as are necessary without one unscheduled stop of production equipment until the war is over and the peace is won, to keep our plant and its equipment in first class operating condition during this emergency period, and to retain it in that condition to meet the postwar period problems—all this to be accomplished with the little critical material available for the upkeep of the machinery and the best use of the few man hours available at the few times allowed for work upon the machinery. If we are able to accomplish this purpose under present war conditions, why not continue the use of maintenance systems and carry on along these same lines into and beyond the postwar period? This should be one postwar plan that we could start during the war.

Before starting to plan a maintenance system, it would be advisable to have an educational campaign; by this I do not mean classrooms, study periods, or the like, but I do mean that the top man in the management of the plant should be the first one to be sold on the fact that it is a desirable system, and if it is not now, it will be a "must" in any plant because a well-maintained plant will have one less postwar problem. He, in turn, must then issue orders making possible and permissible at predetermined times the assignment of operating units to the maintenance crews, equipment that is seemingly in good condition. For it is only by paying attention to what appears to be good equipment that you'll avoid eventually working upon wornout, wrecked equipment. In order to carry on such a program, the management must necessarily recognize results obtained and insist upon following schedules outlined, not only in years when business is good but also in the years when business is bad. For, contrary to our usual teachings, maintenance work must be carried on most intensively in the years when business is bad; it is then that the apparatus can be prepared for continuous, trouble-free service during the periods when business is good.

The departmental supervisor or superintendent is next on a list of executives to be made conscious of and interested in the educational program. Since he is responsible for the successful operation of the department under his jurisdiction, so should he be most interested in learning about and assisting in obtaining better, steadier, trouble-free operations in his department—all this to be gained just by cooperating with the maintenance division. It is his task and responsibility to sell to the operating personnel the idea of reporting the small irregular operations of machinery which, if allowed to continue, eventually will lead to major breakdowns. A report of these irregular operations enables the maintenance men to locate, avoid, or correct conditions that might eventually result in a major breakdown, interrupting production schedules and interfering with delivery to customers of finished products.

The idea having been sold and properly appreciated by the parties who by virtue of their position are able to properly schedule the equipment for maintenance work, the next educational step is to contact the repair foreman and the repair crew. Workmen cannot be too familiar with the machine builder's standards when it is their job to keep these machines up to the builder's standards. All educational information relative to the installed equipment should be available for the maintenance crew's inspection, study, and ready reference. They should also be advised and made conscious of the benefits to them that are derived from planned, orderly overhauling jobs in preference to pushed, rushed breakdown jobs. But don't overlook breakdown jobs as a good source of information when looking for reasons for the failure of machinery. Since a well planned maintenance system makes provision for work being done not only in the years when business is good, but also in the years when business is poor, the workman will be interested in this system because he will be retained on a fairly uniform level of employment.

### Developing Plan for Pump Maintenance

The development of a plan for service or circulating water pumps, moving water without treatment other than that of rough screening, must cover several points. Naturally, the first point to be considered is the number of pumps installed—the number in operation for maximum conditions, the number in operation for normal conditions, and the maximum number in service to meet operations under emergency conditions. What effect, if any, does the temperature of the inlet water have upon the consumption of water and the number of pumps in operation? Will 33 F water permit the operation of fewer pumps than 74 F water? Do fall and spring storms cause turbulent, dirty water that contributes to more than normal wear to moving parts in the pumps? Do we have ready for service replacement parts ready to use while the parts in need of attention are being rebuilt, or do we keep the pump down during the overhauling period? Is it possible to operate with maximum efficiency and take a chance on emergency, or must we be prepared for uninterrupted service at the expense of efficiency?

It is not likely that all the factors can be or will be foreseen the first time we work upon our plan. But with these pertinent facts before us we can start on the pump maintenance system layout. Stormy weather and dirty water indicate

### SPEECHLESS

**Robert B. Harper, vice-president of the Peoples Gas Light and Coke Co., in welcoming the Midwest Power Conference to Chicago, said that when he first arrived in the city he was "speechless" on seeing its wonders (he was six months old at the time, he went on to explain.)**

the loss of essential man hours, critical materials, and waste of money if parts are inspected and necessary replacements and repairs made before the dirty water condition hazard has passed. Reduced consumption of water during the low temperature water season indicates the work should be done just after the storms and before the temperature rises. The need of all pumps for normal and maximum conditions during the warm water season indicates all inspections, replacements, and repairs must be completed before that season. Continuity of service is paramount, so we may overlook some loss in efficiency while waiting for the indicated planned time for overhauling. This condition also indicates the need of replacement parts, parts that will be brought back to manufacturer's standards by shop men while the field man does other work.

This example is representative of what we do to develop plans used in the maintenance of equipment. Feedwater pumps, transformers, transmission lines, oil switches, surface condensers, motor generator sets, deaerators, evaporators, auxiliary motor drives, buildings, cranes, mill machinery, locomotives, diesel electric switchers, and water lines and valves are just a few of the units which are approached in the same general manner in our maintenance planning. In the development of a maintenance plan we should consider, first, known factors influencing operations; second, known factors contributing to outages planned or otherwise. It is well to examine as many old records as are available in order to gather first hand information on this particular part of the plan. These two having been determined, they will be the basis for the next step—first, determine and list work which can be done during low production or offpeak periods, and second, determine and list work which must be arranged for through other groups due to the outage of the unit being worked upon affecting other units.

After the planning is completed and the proper cycles of inspection arrived at, we believe this information in a simplified form should be on a blueprint sheet, say 28 in. x 36 in., placing as many units on the blueprint sheet as possible, allowing room for several inspection date entries for each unit. By looking over a few blueprint sheets prepared in this manner, it is possible in a very short period of time to determine just where we are in our maintenance system plan. The simpler the record system is, the easier the maintenance system will be, but we do need a complete record.

### Records and Log Books

In fact, any maintenance system is greatly enhanced by well kept, accurate records. We believe the most usable of all to be the operating log books. The crew on the turn as well as all equipment in operation is listed at the beginning of each turn, and all changes in equipment, unusual happenings, work performed by operating or maintenance crews, visitors other than plant employees, fuel consumed, material or supplies received or shipped are all entered in the order of their occurrence. Equally important are the daily reports of work done and calls answered by repair and maintenance men, the production of units and condition of equipment by operating crews, including delays and the reason for the delay. These reports and log books are always available for study and checking when any questions of operation or maintenance arise.

No plan would be complete unless it takes into consideration the placing of all replacement parts in first class condition, bringing these

as close to the original or parent manufacturer's standards as possible. We cannot rest at our oars and believe that because we just did a fine overhaul job that we will not need the replacement parts until a distant date; they need immediate rebuilding. Replacement or renewal parts in a maintenance system require exceptional care, not only in handling but also in storage, so that they will be in serviceable condition when required.

No planner should overlook the advantages to be obtained from having sufficient number of tool benches, rigs, special tools, lifts, blocks, slings, receptacles, and other necessary equipment on the job. These few extra details make the job go along smoother, more quickly and efficiently. There will be no looking around or waiting for parts or waiting for someone to finish with a tool before you can use it. In other words, no lost or wasted vital man hours on the job.

## Necessity for Research on Combustion Rates and Ash Behavior

ALTHOUGH pertaining specifically to power boilers, the paper on The Necessity for Research on Combustion Rates and Ash Behavior, by A. R. Mumford, development and research department, Combustion Engineering Co., contained a great deal of information on combustion fundamentals which is applicable to heating boilers as well:

The field of research in fuels is so large that it will be impossible, within the scope of this paper, to more than mention some of the problems concerned with higher rates of combustion. Higher rates of combustion would be beneficial if the reduction in furnace size and cost and in building volume offset the increased costs of operation. The present rates are set by the troubles encountered with ash and to a lesser extent the losses resulting from incomplete combustion. Overall performance figures have been of value but are no longer of great use in solving design problems because the relation of the

average to the maximum is not known or not rationalized. We must know, for instance, how to conclude from the average temperature of the gases leaving a furnace and the furnace geometry what is the maximum temperature existing within the furnace cavity and the point at which it exists. Failures of pressure parts and troubles from intense heat release do not average out over an area or throughout a volume but only over a small part of the area or within a small part of the volume. The average condition is no more indicative of the maximum than the average wind velocity in a county is indicative of conditions in the heart of a cyclone within that county. We must, therefore, develop through research our technique of measurement to permit the advances in theory required before we can improve our design.

**Higher Rates of Combustion**

The present methods of burning solid fuels on grates and in suspension and of burning liquid fuels clearly show differences in intensities which indicate that we are now burning fuels at rates in "hot spots" which if possible on the average would make our furnace volumes comparatively minute. Conversely, if the present averages were maintained and the variations between maximum and minimum rates leveled out, many of the existing "hot spot" troubles would disappear.

The speaker has made tests of boilers fired by pulverized coal in water cooled furnaces in which average heat release rates of 25,000 to 30,000 Btu per cu ft per hr were easily obtained and maintained. Similarly tests of boilers with small refractory lined furnaces fired with oil have been made just as readily with average heat release rates of 175,000 Btu per cu ft per hr. Rumors of tests with release rates with oil fuel over twice this latter value have been heard and direct fired air heaters using oil have been reported as burning millions of Btu of oil per cu ft. Tests made by the writer on stokers burning anthracite coal have shown it possible to sustain rates of combustion as high as 50 lb per sq ft of grate per hr, equivalent to 150 lb or 1,700,000 Btu fired per cu ft of fuel bed per hr. Not all of this heat was released in the bed; the analyses of the gases at the top of the fuel bed showed that some was released by gaseous combustion within the furnace. Considering the fuel bed as a gas producer, producing a gas rich in CO, we can assume that this value should be reduced to about 850,000 Btu per cu ft of fuel bed per hr. The point is that very high rates of combustion are readily obtained with liquid fuels not associated with ash and in beds of solid fuel not disrupted by air flow. The almost fantastically high figures estimated for fuel bed release rates suggest a "hot spot" to average ratio of 30 or 40 to one, which makes us want to question the significance of the corresponding average value for the entire furnace. Little data are available on the variation in the intensity of combustion in the furnace cavity, but no one will deny that large variations exist and, therefore, the average does not indicate the maximum. Our aim should be to find what conditions exist at the "hot spot" in the furnace and how we may increase the volume of the "hot spot" until it fills the furnace and thus bring the average rate of heat release to at least the value which exists at some point in the furnace today, or conversely, how we may bring the indicated maximum intensity down while keeping the average the same.

Part of the problem of higher rates of combustion or the elimination of hot spots is the effective use of the available furnace volume. In many marine boiler tests it was found that the rotary motion imparted to the air by registers on oil burners was dissipated very quickly in rectangular furnaces but persisted in the cylindrical furnaces of a Scotch marine boiler. From this, and other indications, it can be concluded that the shape of a furnace has an important influence on the effectiveness of mixing. It is fair to assume from available observations that some shapes of furnace permit the formation of more eddy currents than others and the initial energy of the fuel and air streams is dissipated by these eddy currents. Furnaces are now shaped to fit available space and to comply with certain obvious requirements of practical and economic construction. From considerations of the change of kinematic viscosity of gases with temperature it seems likely that we must be prepared to use energy much more freely in large furnaces to secure the same degree of turbulence as we do now in small furnaces. This relation of the factors controlling the ability to secure thorough mixing of the fuel and air for combustion is at present entirely on an empirical basis and should be rationalized by research.

Oil fuels are being burned in furnaces under pressures of 5 or 6 in. of water quite generally, and we may learn, after the war, that pressures measured in pounds have been used. Increasing the number of oxygen molecules within a given volume by increasing the furnace pressure ought to increase the opportunity of collision with fuel particles and, therefore, the intensity of combustion. The answer may be partly known and may be made available later, but even though the answer has been found for liquid or gaseous fuels, its application to solid fuels will probably be subject to modification because of the ash content. The British Coal Utilization Research Association in a report states that "By measuring the energy of its interaction with methyl alcohol it was established that coal substance is so finely structured that the surface accessible to oxygen molecules ranges from 100,000 to more than a million sq ft per lb according to its rank." This seems large but is probably small if compared to well atomized oil. The problem, therefore, is not essentially one of pro-

viding contact surface in the fuel but of utilizing more effectively the contact surface provided by nature. The more effective utilization of this contact surface may come about by crowding more oxygen molecules into the small spaces, i.e., by increasing pressures in the combustion chamber. The extra cost of the structural changes necessary to resist these pressures will be at least partly offset by the decreased size of the furnace. At any rate, full scale experiments are not economically feasible and small scale laboratory work should be undertaken.

All of us who have burned anthracite have been interested in, and perhaps annoyed by, the particles carried out of the furnace in suspension in the gases. Some of us have tried to burn these particles and found it difficult. The BCURA

**EXPERT DEFINED**

**H. B. Dirks of Michigan State College defined an expert as "one who is away from home," in his introductory remarks at the opening session of the Midwest Power Conference.**

points out that "The marked decay in reactivity shown by anthracite when heated had been investigated. This property renders the relighting of partly consumed anthracite peculiarly difficult. The effect was traced to a loss of 'inner' surface (accessible to oxygen) when the fuel is heated above 800 C." If the reactivity of the unconsumed portion of a solid fuel decreases during the burning process, what is the extent of the decrease and what factors affect it? It seems reasonable to expect some decay in reactivity of the fuel as it burns because the most reactive portion would naturally be the first consumed. Some change of this character may account for the necessity of assigning a large part of the volumes of existing furnaces to the combustion of the last portions of solid combustible in comparatively quiescent zones. In such zones mixing action is at a minimum and probably the reactivity is low, both factors working against the reduction in furnace volume. Perhaps when measurements of reactivity are made on representative samples of the fuel in the progress of its combustion we will be able to apply energy for mixing as needed and thus maintain "hot spot" burning rates in what are now quiescent zones of the furnace.

In any consideration of combustion in fuel beds a very obvious point is that the fuel is in large enough particles so that it is not floated on the air stream. Under such conditions the scrubbing action of the air against the surface of the incandescent fuel is a maximum and reaction is rapid. Much work on fuel beds has been done. Henry Kreisinger did some pioneer work on combustion in fuel beds and pointed out that all oxygen would be consumed from the air passing through a bed of incandescent fuel 4 in. or more in thickness. The significance of this work on fuel beds is that it is theoretically and practically impossible to supply sufficient air for combustion through those sections of the fuel bed composed of a thickness of 4 in. or more of uniform incandescent fuel. Obviously the deficiency in air must be made up through other portions of the fuel bed which are not completely ignited or which are burned out. The streams of gases deficient in oxygen must be mixed with those rich in oxygen either by providing time and volume enough for diffusion, or by compulsion in the form of arches. Arches can work perfectly only at the rate of combustion or flow of gases for which they are designed. At other rates some assistance must be given in the form of steam or air jets to provide the mixing. In 1924 the speaker published a study of combustion of anthracite on stokers in which he used steam jets to simulate the action of a rear arch. The results, using steam at 150 psi, were approached but not equaled by the arch. If air jets are used, the air must have sufficient energy to penetrate the gas stream and supply the needed oxygen. The spur for research along the lines of combustion in fuel beds is the high volumetric heat release rates obtainable in fuel beds and the comparative ease with which high release rates may be obtained in the combustion of the gases rising from the bed. Obviously the desirable condition is the maximum turbulence of the air and gases passing through the fuel bed obtainable without disruption of the bed and then energetic mixing devices and air supply in the furnace above the grates to insure rapid gaseous combustion. Perhaps some combination of combustion on grates, in suspension, and gaseous combustion will be indicated as the most economically efficient use of available volumes.

In the study of combustion with emphasis on the factors affecting higher rates, the rate of loss of heat from the flame is of importance. Most of our large units are designed with some arrangement of water cooling at the envelope of the furnace and this cold surface absorbs heat as a function largely of the emission temperature and the black body volume of the flame. Except in furnaces having a high degree of turbulence the convective component of absorption is small. At present it is quite generally the practice in estimating the emission temperature of furnace gases to assume it to be the same as the average temperature of the gases leaving the furnace cavity. This assumption is safe enough for a furnace which does not approach limiting conditions at the "hot spot," but is questionable when it is known that wide variations exist within the furnace cavity and the maximum may subject the cooling means at some point in the envelope to a strain beyond its limits. Here again the relation of the temperature of gases leaving a furnace to the emission temperature of the gases within the furnace should be studied with emphasis on the maxima and measuring devices are needed. John Blizard pointed out in his recent paper on Absorption of Heat by the Walls of a Furnace before the ASME, that the amount of heat absorbed in a cooled fur-

nace was found to be related to the rate of firing. Under his conditions most of the heat was transmitted by radiation and his findings are of interest, therefore, in indicating that the black body volume of the flame changes with the rate of fuel consumption. Temperatures of the flame at the different rates varied somewhat because of minor variations in $CO_2$ but not sufficiently nor consistently enough to explain the variations in the quantity of heat absorbed. Although the trend in large utility units seems to be toward operation at nearly constant loads, a sufficiently large number of units will be needed for operation at varying loads in industry to make the determination of the relation of black body volume of flame to fuel consumed and to furnace geometry an important research project.

If research rationalizes the manner in which we may obtain higher liberation rates and more effective use of available volumes, it must also solve some related problems having to do with the design and cooling of the furnace envelope. Paradoxically the liquid phase of water has been shown to be less effective as a cooling medium than the mixture of liquid and vapor phases at a slightly higher temperature. Some work on the transmission of heat to boiler tubing at high pressures reported the empirical conclusion that the film resistance under evaporating conditions was negligible as compared with the resistance of the metal of the tube. In the absence of vapor the film resistance was found to be appreciable. The low film resistance is believed to depend on the disruption of the liquid film during the formation and escape of vapor bubbles. Assuming the relative unimportance of the film resistance under evaporating conditions as compared to the resistance of the metal of the tube, Dr. Jakob computed the tube surface temperatures at certain rates of heat absorption and thus confirmed the conclusion. Assuming then that the furnace cavity will be cooled by evaporation, how can the resistance due to tube wall thickness be reduced, particularly at high pressures other than by decreasing the tube diameter? In long small tubes, practicable in forced circulation boilers, the sum of the radiation and convection components in a furnace having an extreme degree of turbulence and a high rate of heat release may produce tube surface temperatures 150 F above saturation. Ely and Schueler have reported the failure of a thick wall tube due to temperature stresses alone even with adequate circulation of the cooling medium, indicating that, at that location, the stress produced by the temperature differential through the metal exceeded the resistance of the metal to stress and the rate of heat transmission had to be decreased. What are the temperature drops through available alloys and what is the relation of these drops to those found for carbon steel of the same strength? As our operating pressures approach the critical it is apparent that the amount of heating surface devoted to evaporation will decrease and the proportion devoted to water heating and superheating will increase. At the critical pressure no evaporating surface is required because of the absence of latent heat and the boiler will then consist of a water heater and a superheater. It should be an interesting problem to determine the disposition of heating surface so that the change in the cleanliness factor of the furnace will not upset the designed balance of the heating surface or will not completely preclude operation at more than one load. The amount and distribution of the heating surface must be based on a knowledge of the rate at which heat is released and available, as well as the mechanism by which it is transmitted to the liquid. At low steam pressures, the ability of a tube to absorb heat was seldom, if ever, questioned, but now we need to know more of the limits because we are approaching them.

It is generally recognized that ash is always a nuisance and sometimes constitutes an operating problem. The degree of the nuisance or problem differs with the quantity or the physical properties of the ash and sometimes with both. The ashless fuels, oil and gas, do not constitute a sufficiently large proportion of our fuel reserves to warrant any optimism that they will remove, by substitution, the obstacle that ash now constitutes in obtaining higher rates of combustion. Furthermore, 75 per cent of our solid fuel resources are in the central and western fields, which are generally likely to contain appreciably more noncombustible material than the eastern. It seems obvious that any problems introduced into the burning process by the presence of noncombustible material are likely to increase as our resources decrease. For this reason and because the ash in coal is now much more of a problem than an annoyance, research should be directed toward the amelioration of the problems it introduces.

### Ash Behavior

The two principal problems introduced into the combustion of solid fuels by ash are the insulation of heat absorbing surfaces and the obstruction of gas passages. Heat absorbing surfaces are insulated by the adherence of particles of ash forming a layer which will vary in thickness in accordance with the physical properties of the ash. The more refractory the ash, the less the trouble to be expected, assuming that the present laboratory measurements of softening and fluid temperatures are indicative of the viscosities involved. The result of this insulation is to disturb the design balance between evaporating and superheating surfaces and this necessitates changes in operating procedure to restore the desired conditions. When the layer of ash deposits on adjacent boiler tubes between which the gases are flowing the layer may build to a thickness which will materially increase the draft loss or completely obstruct the flow, necessitating reduction of rating until

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the obstructions are removed. Minor problems are introduced by the variation of the relation between viscosity and temperature affecting the removal of ash as a fluid from furnaces.

Ash also has some chemical properties. We believe that the chemical properties of the ash are responsible for the loss of metal from furnace tubes under the conditions of gas composition existing in parts of the furnace. This particular reaction is being studied in the laboratories of the Bureau of Mines in Pittsburgh under a cooperative agreement. When this investigation reveals the fundamental reactions involved, the control of the reactions will be apparent.

The pioneers in the field of ash softening, fusion, and liquid temperatures are today in the forefront of those recognizing the need for new studies of the physical properties of ash. Much work has been done but more is necessary to relate laboratory measurements to field conditions. At present certain physical properties can be predicted from a knowledge of the chemical composition of the ash, but until more is known of the actual conditions to which the ash is exposed in the furnace, the greater part of the value of this information will remain potential. If, however, we determine the conditions of exposure and relate them to the geometry of the furnace and the firing rate and method, it is not too optimistic a prediction to say that we ought to be able to estimate the location and thickness of deposits and infer the resultant effect on heat transfer from our knowledge of the black body volume and the emission temperature of the flame and the physical properties of the ash. We are uncertain at this time whether the conductivity of slag varies with its chemical composition, but a study of this property is underway at the Bureau of Mines under a cooperative agreement. The results of this study will indicate whether we may eliminate chemical composition from consideration in determining the insulating effect of the layer of slag and consider only its thickness and physical structure. The effect of physical structure will be studied by relating the laboratory results to field conditions. The thickness of the deposit will obviously vary with the conductivity of the ash; thus an ash having a high resistance to the flow of heat will become fluid at the surface in comparatively thin layers. To relate the laboratory work to field conditions we must measure the heat flow by existing temperature drop methods and develop instruments to measure the furnace side surface temperature of the slag layer. Without a more complete knowledge of the physical properties of ash any attempt at eliminating it as one of the factors preventing the attainment of higher rates of combustion must be almost entirely empirical and probably costly. With more knowledge the designer can make some provision in his design to fit it to the predicted conditions.

It is known that the combustible substance of coal is in general less dense than the noncombustible substance. Separations have been made by flotation and the heavier fractions found to contain the larger proportions of noncombustible material. Coal washing is practiced at some mines, but only where the ash content of the original product was considerably higher and more troublesome than competing coals. Some users have justified the cost of cleaning by reduction of operating costs in existing installations. There is evidence also that the cost of cleaning is not justified when the furnace can be designed to handle the uncleaned fuel. At any rate, more knowledge of the properties of the ash will permit the formation of conclusions with more assurance.

### Conclusion

We have endeavored to point out the necessity for fuel research and the lines it might take. To accomplish two general objectives, (1) the increasing of the combustion rates, and (2) the amelioration of troubles from ash, we must develop and use instruments to measure local conditions within the furnace cavity or fuel bed, and the physical and chemical properties of the ash. We should no longer predicate extrapolated performance on existing average conditions, but should study the maxima in detail. Dr. Carl B. Boyer in his History of the Measurement of Heat said, "As science goes on in the search for still greater unity, let it be remembered that the way is paved, not with unmeasured speculation, but with the objective data of patient quantitative research."

## Corrosion Inhibitors in Condenser Tube Alloys

CORROSION Inhibitors in Condenser Tube Alloys was the title of a paper by Austin R. Zender, general sales director, and C. L. Bulow, research chemist, of the Bridgeport Brass Co., which was presented by Mr. Bulow. It is given here without the tables and curves shown on the screen during its presentation:

Corrosion may be stifled by the addition of a corrosion inhibitor either to the alloy or to the corrosive solution. Inhibitors added to the solution generally fall into two broad classes—organic and inorganic. Each type has its own peculiar merits or shortcomings, depending upon the circumstances of its use. Corrosion inhibitors may function primarily by screening or blanketing (polarization) of the cathodic or anodic area. Many substances can operate either as inhibitors or accelerators of corrosion, depending upon the metals and solutions involved. The in-

organic type of inhibitors (salts) seem to vary most in their behavior since a given salt which may act as an inhibitor towards one metal may act as an accelerator towards another.

Visual examination of steel corroding in dilute sulphuric acid reveals that hydrogen evolves from a relatively small number of points. The hydrogen forms a blanket or screen on most of the cathodic (noncorroding) surface, and at a few points coalescing of hydrogen atoms or hydrogen molecules takes place with the formation of hydrogen gas which breaks away from the surface at these active points. This suggests that it would require relatively little blanketing by some particles which could be made to cover these points, which in turn might stop the liberation of hydrogen gas. Actually, in practice, the addition of a small quantity of a basic nitrogen compound is enough to stop the evolution of hydrogen and such organic inhibitors have been used for years in the pickling of ferrous metals. Generally these inhibitors reduce the rate of removal of scale from the metal only slightly and quite effectively protect the base metal. For example, a hot 10 per cent sulphuric acid solution (by volume) at 150 F was found to attack a low carbon steel at the rate of 180 milligrams per square decimeter per hour. The addition of 0.05 per cent (by volume) of a commercial inhibitor lowered the corrosion rate of this steel to 9 milligrams per square decimeter per hour. Here the inhibition apparently depends upon the movement of positively charged particles toward the negatively charged cathodic points where they are adsorbed by the negatively charged surface of metal where they hamper the cathodic reaction and thereby decelerate the rate of corrosion.

Generally colloidal particles are positively charged in acid solutions and are therefore more effective in acid solutions than in alkaline solutions. Colloids in alkaline solutions are negatively charged and move towards the anode. Such a condition may lead (by differential aeration) to localization of the attack, i. e., to pitting. It is difficult to find suitable organic inhibitors for use in neutral salt solutions, such as sodium chloride, since corrosion in these solutions may lead to an alkaline reaction around the cathodic area. In soft waters or in buffered salt solutions, alkalies cannot form very readily at the cathodic points, consequently the organic inhibitor behaves in a dependable manner.

### Effectiveness of Nitrogen Compounds

On the assumption that nitrogeneous inhibitors (aromatic amines) are attached in acid solution to the metal through the nitrogen atom Mann, Lauer and Hultin[1] showed that those compounds which are so shaped as to be likely to screen large areas are the most efficient. The amines which they tested dissociate in acid solutions to yield a positively charged inhibitor (i. e., cathodic inhibitor). These positively charged particles (ions) are considered as protecting "metals against acid corrosion by forming a blanketing layer that is adsorbed on the metal surface through the nitrogen atom."

Pure chemicals such as certain sulphur and nitrogen compounds (pyridine, amines, urea, quinoline, ethiodide, benzyl, thiocyanate, etc.) are very effective but too costly for widespread use. For example the addition of 0.05 per cent benzyl thiocyanate to 10 per cent sulphuric acid solution very effectively inhibited water line corrosion of copper base alloys.

### Proprietary Inhibitors

Many of the proprietary inhibitors are quite efficient and relatively inexpensive. Some of these inhibitors are byproducts from industrial processes such as sulphonated wastes, glue and gelatine from meat packing plants, acid sludges and extracts from coal tar products, and sulfide cellulose lye from the paper industry. Since the above mentioned materials and other substances, such as egg albumen, dextrin, oil, agar, tannin, and gum tragacanth possess certain corrosion inhibiting properties, it has been suggested that it is likely that unidentified inhibitors occur in foodstuffs which may account for the comparatively long life of cooking vessels (copper, aluminum, etc.). It is also likely that certain waters contain corrosion inhibiting colloids, since some nonscale forming waters are not as corrosive as expected. It is known that some waters carrying suspended clay and other materials are less corrosive than similar waters containing very little foreign matter.

### Oils as Inhibitors

Emulsifying oils are sometimes added in small quantities to cooling waters with the formation of a milky liquid which produces appreciably less corrosion than the untreated waters. The use of emulsified oils does not seem to have a very detrimental affect on heat transfer even though the film of oil which forms on the metal surface seems to be practically continuous. The main objection to the widespread use of many oils is the injurious action, of excessive concentrations, on the rubber parts of cooling systems.[2]

### Inhibition by Salts

Inhibitors of the inorganic type (salts) usually function by stifling the anodic reaction instead of the cathodic reaction. These inorganic inhibitors operate in a manner similar to that of the organic inhibitors which have been briefly discussed. That is, the salts dissociate to give negatively charged anions which are attracted to the positively charged corroding anodic points. The chromates, silicates, phosphates, carbonates, arsenates, and nitrites have been used. The chromates apparently are widely used to prevent corrosion in many metallic systems handling recirculating cooling water or brine solutions such as automobile radiators,

air conditioning equipment, coolers, etc.

### Arsenic and Antimony as Corrosion Inhibitors

In 1908, Dwight[3] reported that arsenic trioxide was an effective corrosion inhibitor (protected steel jackets) when added to cooling water contaminated by sulphuric acid from smelter fumes. Watts[4] and Clarke[5] and others have reported on the effectiveness of arsenic and antimony compounds as corrosion inhibitors. In 1924, Bengough and May[6] described an experiment pertaining to the effect of arsenic trioxide as an inhibitor of copper deposition on 70-30 brass in cupric chloride solution. Their experiment consisted of suspending pieces of condenser tubing in beakers containing 150 cc cupric chloride solution and known amounts of arsenic trioxide. It was found that arsenic was effective in preventing the deposition of copper when the ratio of the concentration of arsenic to copper in the solution was approximately one or more. They concluded that "In view of the remarkable effect of arsenic in influencing the nature of the corrosion products of brass, and through them the corrosion rate of the alloy itself, it was decided to examine the influence of a number of metals which commonly occur in brass, either as accidental or intentional additions."

### Mechanism of Dezincification

In the absence of a dezincification inhibitor either in the corrosive solution, or in 70-30 brass, the corrosive reaction would proceed with the brass dissolving as an entity followed by the redeposition of copper (instead of hydrogen) at the cathodic area. Abrams[7] pointed out in 1922 that a relatively high concentration of copper ions is an important prerequisite to dezincification of brass. Dezincification is also favored by (1) a limited supply of dissolved oxygen, (2) the presence of certain salts, (3) a low water velocity or stagnation and (4) an elevated temperature since these factors aid in increasing the possibilities of obtaining a high concentration of copper ions. Under certain conditions, the copper redeposited may form a continuous layer, or when foreign material settles in or becomes attached to the walls of the brass tube or pipe, (which leads to a non-uniform distribution of oxygen) the corrosion instead of appearing as pits, appears as plugs of redeposited copper. For example, muntz metal piping used in long horizontal runs carrying hot water has failed from plug type dezincification localized in the lower half of the pipe. With the pipes mounted in a vertical position, the copper plugs are scattered in a random manner.

### Effectiveness of Other Inhibitors Compared With Arsenic

When Bengough and May[6] discovered that 70-30 brass condenser tubes containing a few thousandths of 1 per cent of arsenic did not dezincify, they modified 70-30 brass by addition of 1 per cent tin, 0.5-1.0 per cent nickel, 0.5-2.0 per cent aluminum, 0.5 per cent tungsten, 0.04-2.0 per cent manganese, 0.48-0.91 per cent of iron and 0.01-.04 per cent arsenic. Their work showed that otherwise pure 70-30 brass containing as little as 0.02 per cent arsenic would not dezincify; that is, the arsenic would prevent "copper deposition in the heart of the metal in any conditions likely to occur in sea water up to a temperature of at least 50 C and that even 0.01 per cent arsenic may be effective." The effect of 0.02 per cent arsenic in preventing dezincification was reported as being more effective than 1 per cent tin. The order of the elements preventing dezincification was as follows: Arsenic, tin, nickel, tungsten and aluminum, with manganese and iron increasing the rate of dezincification under these conditions. In the same year, Nixon[8] reported that brass containing less than 0.3 per cent of arsenic was remarkably resistant to dezincification.

Since 1924, F. W. Fink,[9] F. M. Barry,[10] W. Lynes,[11] J. H. Hollomon and John Wulff[12] have reported on the effectiveness of arsenic as a dezincification inhibitor. During this interval, it was also learned that other elements in group 5 of the periodic table, namely, phosphorus and antimony, were also effective in inhibiting dezincification. Barry[9] reported the results of a total immersion test conducted in 1 per cent cupric chloride solution. The effect of the dezincification inhibitors, arsenic, antimony and phosphorus, at 80 F was very marked when compared to regular admiralty metal. All three inhibitors (0.03-0.06 per cent) had about the same effect. However, it was reported that the arsenic inhibited alloy was not as effective as the antimony or phosphorus at 125 F due to a peculiar intercrystalline corrosion which appeared in the arsenic admiralty. This is unusual since no service failures of this type have been encountered. Subsequent work by Lynes on the same inhibitors in 70-30 brass and admiralty metal failed to reveal any such variation in the behavior of the arsenical admiralty metal when tested in a 5 per cent cupric chloride solution containing 5 per cent hydrochloric acid. "Conditions were established in the laboratory that resulted in complete dezincification of 0.350 in. diameter plain 70-30 brass and admiralty metal rod in a period of 10 weeks. Under these conditions, the same alloys modified by about 0.03 per cent arsenic, antimony or phosphorus did not dezincify; neither did corrosion of the modified alloys progress along grain boundaries." It seems that the intercrystalline corrosion (not cracking) reported by Mitchell[13] and Barry in their annealed specimens of arsenical admiralty is due to a factor or factors which are not at present known.

### Mechanism of Inhibition

These numerous investigations show that the addition of arsenic to brass definitely hinders the redeposition of copper. It has been recognized that arsenic appears as a very thin black or grey colored film on the corroding brass. For example, a few minutes immersion of an arsenical brass in a 50 per cent nitric acid solution (specific

gravity 1.20) will produce a relatively thick black film. This black film can also be produced on arsenical brass by electrolysis of an arsenical brass anode in 5 per cent sulphuric acid solution using a carbon cathode and a low voltage direct current (8 volts). During the process of electrolysis a black film gradually forms and increases in thickness. As the thickness of the film increases the resistance of the film increases. This increase in resistance of the film leads to a decrease in the amount of current flowing through the cell until the current drops to a small fraction of the initial value. This low current continues to flow until the black film becomes thick enough to peel off, and then the current jumps back up to the original high current value. This cycle repeats itself until the anode is dissolved. In the absence of arsenic this phenomenon is not so strikingly brought forth and solution of the anode therefore takes place more rapidly.

Hollomon and Wulff[14] examined by x ray and electron diffraction methods the black colored corrosion film developed on the corroding surfaces of arsenical 70-30 brass and antimonial 70-30 brass and concluded that brass containing approximately 0.03 per cent or more of arsenic becomes coated with redeposited arsenic which is subsequently oxidized to arsenic trioxide. For the antimonial brass the mechanism is practically the same, but the redeposited antimony does not oxidize quite as rapidly. "Therefore, the increase in potential is primarily due to the formation of the arsenical film and the subsequent increase of pore resistance." That is, the potential of the alloy shifts toward a more noble value which approaches that of copper. Thus it seems that the mechanism of the prevention of dezincification of brass by arsenic is quite clearly explained.

### Effect of Inhibitor in Sea Water

It is evident that the laboratory dezincification test in cupric chloride is very useful for exploratory work. It is, however, also true that such solutions do not simulate service conditions closely enough to give a dependable indication of what may be expected when inhibited brasses are exposed to various cooling waters, such as those coming from wells, lakes, rivers and the ocean. Consequently, tests were conducted in flowing sea water, through the courtesy of the International Nickel Co. at its sea water testing station at the plant of the Ethyl Dow Chemical Co., Kure Beach, N. C. The specimens used were in the form of ASTM sheet tension specimens 0.050 in. thick with holes drilled in the ends for fastening the specimens on bakelite insulated rods using bakelite spacers.

The results obtained indicated that the addition of 0.04 per cent of arsenic to admiralty metal aids in inhibiting dezincification in sea water and that it also lowered to a small degree the overall corrosion rate. The low ratio of loss in tensile strength to loss in weight for arsenical admiralty metal clearly indicated that no intergranular corrosion or dezincification occurred during the first year in sea water.

The addition of 0.1 per cent of arsenic to muntz metal very markedly inhibited dezincification in sea water. The presence of 0.10 per cent of arsenic had an effect almost equal to that of 0.75 per cent of tin. Examination of the broken tension specimens revealed that the high ratio of loss in tensile strength to loss in weight (3.88) for plain muntz metal was due to deep uniform dezincification.

The ratio of depth of penetration calculated from loss in tensile strength to depth of penetration calculated from loss of weight is used to reveal the presence of cracking, intergranular corrosion, pitting, dezincification, or incomplete removal of corrosion products. Uniform corrosion is indicated by ratio values which approach unity (1.0).

### Effect of Inhibitors on Stress Corrosion

In addition to the sea water corrosion test, a group of 17 alloys containing varying amounts of phosphorus, antimony and arsenic were subjected to stress corrosion tests in an atmosphere of moist ammonia and air. The result of this work indicated that the stress corrosion cracks produced were in the main transcrystalline and that no marked difference could be found between all of these alloys. These results have received further substantiation from service records. When cracking occurs in all of these alloys generally the cracks in service are of a transcrystalline nature resulting from stress corrosion, season cracking, or corrosion fatigue. Occasionally intercrystalline cracks occur and these cracks are found in regular admiralty as well as admiralty containing phosphorus, arsenic, or antimony. Intercrystalline cracks develop in regular and inhibited admiralty metal when stressed (applied or residual) in contact with solutions of mercury salts or with liquid metals, such as low melting point solders or mercury.

### Conclusions

After many years of experience with the use of arsenic in brass to inhibit dezincification, May[15] reported: "The behavior of aluminum brass can also be varied in an important manner by adjustment of its arsenic content. In the absence of arsenic this alloy, like 70-30 brass, has a tendency to undergo dezincification, but this tendency is completely removed by a suitable small addition of arsenic (e.g., 0.02 per cent), without in any way affecting the resistance of the alloy to impingement attack. An even higher arsenic content may be used if the tubes are required especially to resist deposit attack; an aluminum brass with 0.04 per cent or 0.05 per cent arsenic resists deposits of sand far better than any known condenser tube alloy, with the possible exception of those with a very high nickel content. The importance of the control of arsenic content of aluminum brass is

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widely realized, and it has long been the practice of some tube makers to add about 0.02 per cent arsenic in order to give resistance to dezincification. The author is of the opinion that generally the addition of a small percentage of arsenic to aluminum brass is a decided advantage. It must, however, be borne in mind in this connection that in those cases, probably rare in marine condensers, where resistance to the effect of water containing $H_2S$ is required, a tube which undergoes slow dezincification may have a longer life."

It has been shown by reference to previous experimental work done by others and the author's work that phosphorus, arsenic, and antimony inhibit or prevent dezincification and lower the rate of corrosion of alloys containing these inhibitors. More information is available in connection with arsenic as a dezincification inhibitor than the other inhibitors. Millions of pounds of arsenical admiralty metal and aluminum brass (containing arsenic) condenser tubes have entered service during the past 10 years in the interest of longer tube life. This does not include English experiences, where arsenical brasses have been used for a much longer time.

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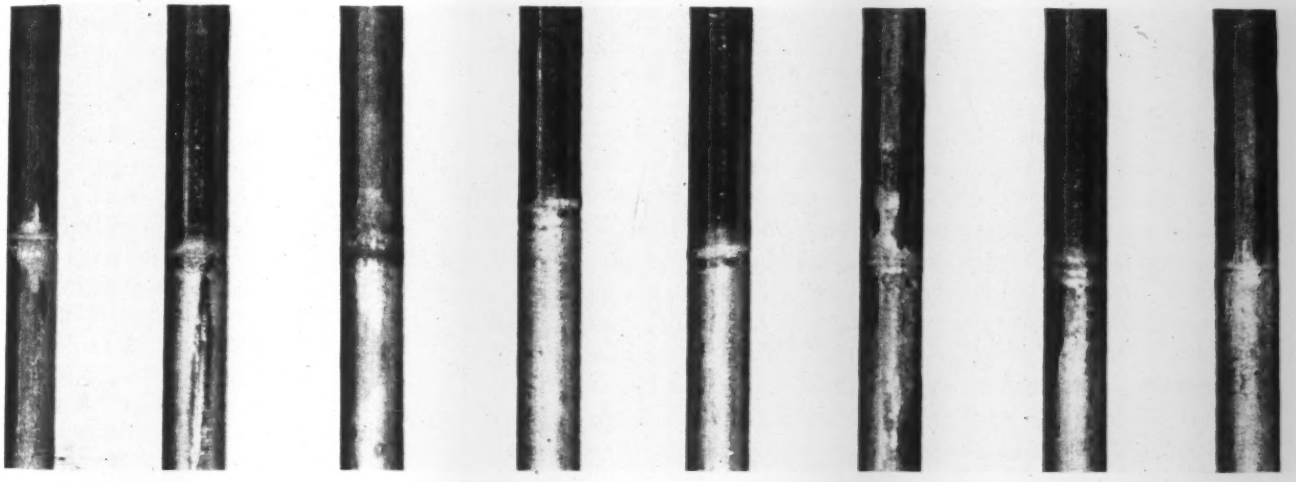
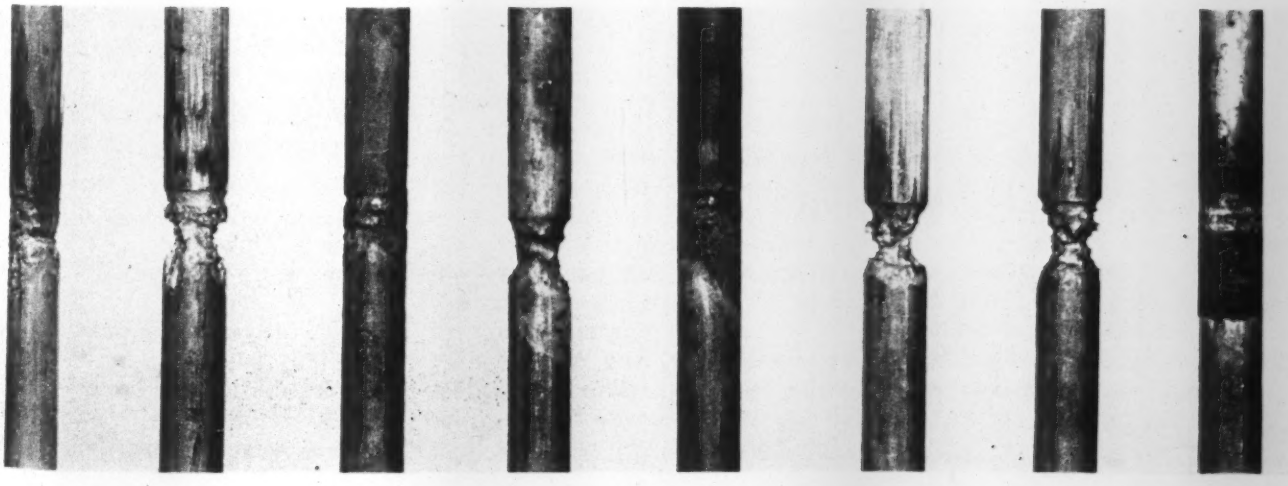
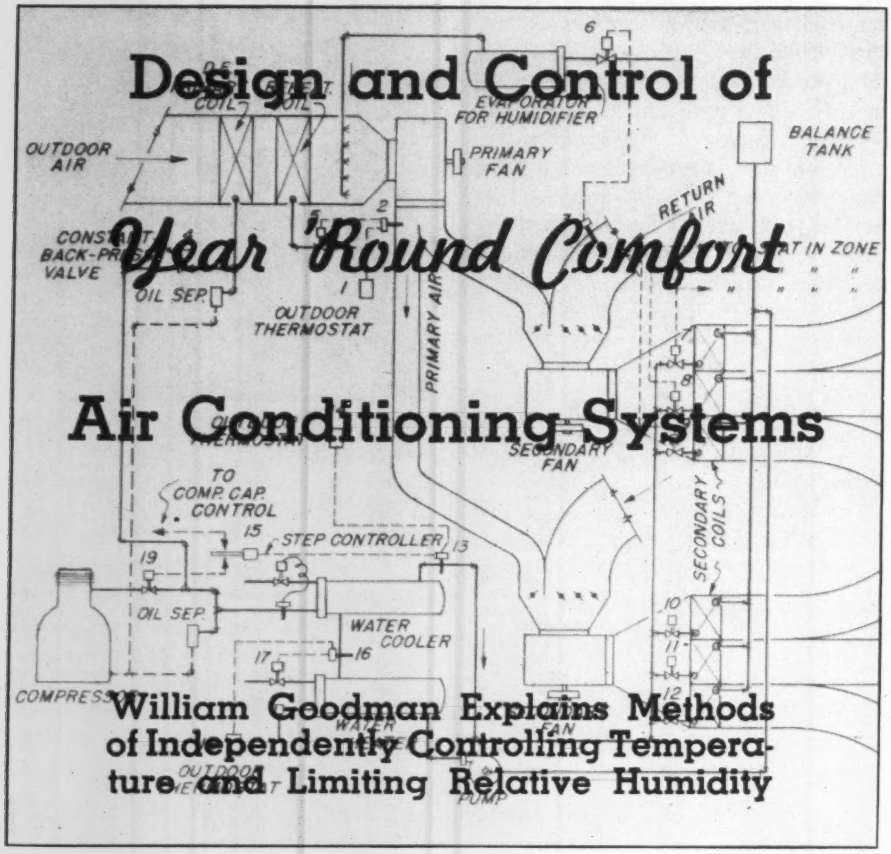




**These illustrations show the difference in the extent of attack occurring in inhibited (above) and non-inhibited (below) sulphuric acid solutions**

# Design and Control of Year 'Round Comfort Air Conditioning Systems




## William Goodman Explains Methods of Independently Controlling Temperature and Limiting Relative Humidity

**The basic principles that govern the design and control of year 'round conditioning systems are organized and discussed in a systematic manner in the series of articles of which this is the third. No attempt is made to show a collection of control and equipment arrangements except insofar as they are needed to illustrate the various points discussed. Instead, the methods of accomplishing certain desired results in air conditioning systems and the broad basic principles that govern the applicability of different types of conditioning systems and controls are presented from a refreshingly new viewpoint. Those who are now designing and will be designing comfort air conditioning systems for postwar installation will find this discussion timely and practical, for no matter what the developments of the future, basic principles will always govern. Mr. Goodman is consulting engineer for the Trane Co., and is a member of Heating, Piping & Air Conditioning's board of consulting and contributing editors.**

IT SHOULD be obvious that the most desirable air conditioning system for year 'round use is one in which the temperature and humidity can be controlled independently of each other at all times. In the simple systems previously described, there was no independent control of temperature and humidity. If temperature is controlled either by starting and stopping the compressor or by reducing the volume of air, the relative humidity in the conditioned space is likely to rise during periods of light heat load. On the other hand, if during periods of light heat load the compressor were run long enough to lower the humidity or if a sufficiently large volume of air were supplied, the dry bulb temperature would be lowered too much—resulting in a cold room with a high relative humidity. These conditions of high relative humidity during periods of light heat load can be avoided only by the use of a system in which the humidity and temperature can be controlled independently of each other. Thus, it should be possible to maintain the temperature at any desired point with a small heat load, and at the same time to limit the relative humidity to some predetermined maximum level.

Although two relatively simple methods of control have been described—that is, on and off operation of the compressor and reducing the air volume through the dehumidifier—these methods (as has been shown) are not satisfactory for a year 'round conditioning system. Nevertheless, these methods, in one manner or another, are frequently used in even large and complex air conditioning systems. If one digs beneath the complex control system and the equipment layout, it is amazing how often the control of dry bulb temperature will be some variation of the on and off system or the method of reduced air volume through the dehumidifier. Thus, in many large systems the temperature is controlled by throttling the chilled water flowing through a coil or air washer. The effect is the same, although not quite so drastic, as having the compressor or the water turned on and off in response to the dry bulb temperature. Some variation of reduced air volume through the dehumidifier is also frequently used in large systems—quite often the bypass. Many an air conditioning system, when reduced to its bare elements will be found to consist of either control of refrigerant or control of air volume from dry bulb temperature. The results in such systems will be about the same as the results previously described for the simple, elementary on and off or reduced volume systems.

### Independently Controlling Temperature and Limiting Humidity

For year 'round air conditioning, and especially for the in-between seasons, independent control of temperature and humidity is essential. By control of humidity is not meant the maintenance of an exact humidity but means for preventing the humidity from exceeding some predetermined limit during the spring, summer, and fall. There

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are but two methods by which temperature and humidity can be controlled independently of each other under all conditions: (1) cooling the air to the dew point temperature required to maintain the relative humidity and then reheating the air sufficiently to maintain the desired dry bulb temperature, and (2) dehumidifying the air by chemical means, such as silica gel or lithium chloride, to the dew point temperature necessary to maintain the desired relative humidity, and then recooling the air sufficiently to maintain the desired dry bulb temperature. As a very general rule, the first method is likely to be lower in first cost. The second method, unless cold well water is available, usually requires in addition to the chemical dehumidifier the installation of a refrigerating system large enough to take care of the sensible load. However, there are occasions where operating costs are such as to justify the installation of a silica gel or lithium chloride plant for dehumidifying the air together with a refrigerating plant for handling the sensible load. In most installations the first cost is likely to be lower for just the refrigerating plant and reheating.

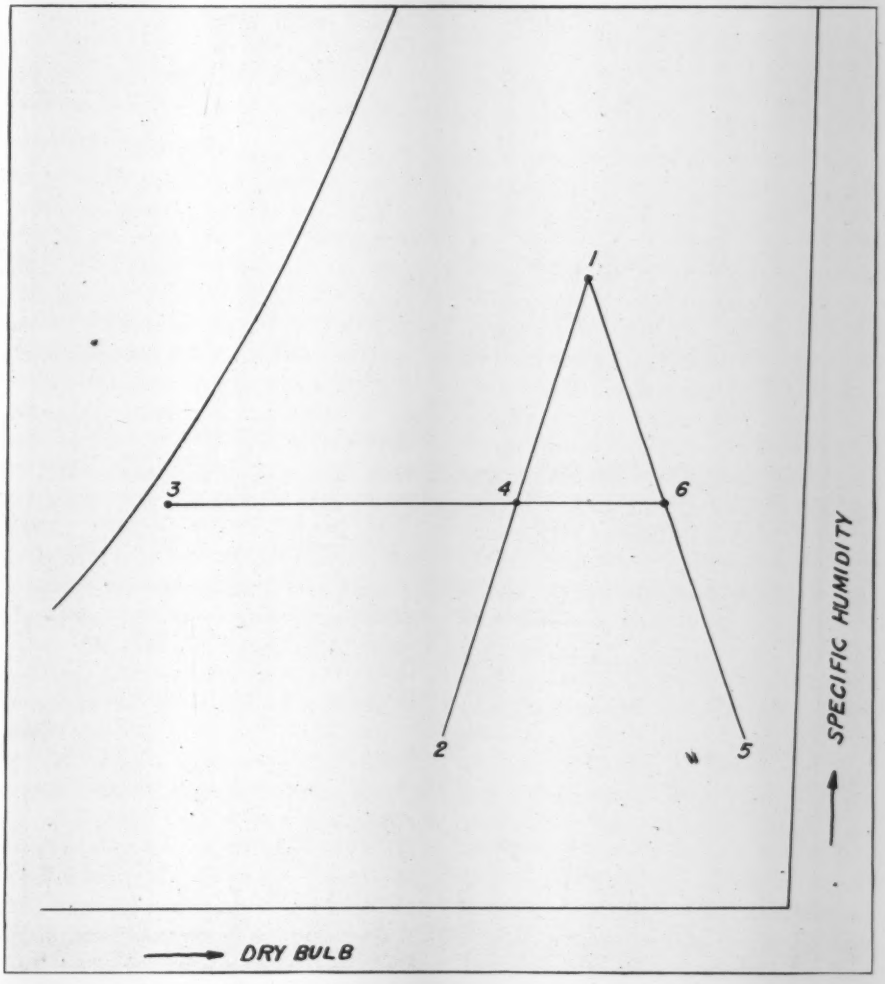
Referring to the skeleton psychrometric chart of Fig. 3, it is obvious that regardless of the location of the ratio line, the air can be cooled and dehumidified to any required dew point and then reheated to any dry bulb required by the ratio line. Thus, for a ratio line like *1-2* the air can be cooled and dehumidified down to a point like *3* and then reheated to point *4*. Or if the ratio line is like *1-5*, the air, after being cooled and dehumidified to point *3*, can be reheated to point *6*. Only if air in the condition represented by point *6* is delivered to the conditioned room can the room temperature and humidity represented by point *1* be maintained for a ratio line like *1-5*.

Either cooling and dehumidifying with reheating, or dehumidifying chemically and recooling, offer means of obtaining completely independent control of temperature and humidity. With such systems, regardless of the sensible load or even of the moisture load in the conditioned spaces, any desired temperature and humidity within the capacity of the equipment can be maintained regardless of the season of the year. These two methods—dehumidifying and reheating, and dehumidifying and recooling—are the only two methods known today by which completely independent control of temperature and humidity can be obtained. For winter, of course, some means of adding humidity must be provided, but this is relatively simple and ordinarily presents no problem. Methods and controls for humidifying in winter will be discussed in more detail later.

Of course, in comfort conditioning, as has been previously stated, exact control of humidity is not required. However, it is important that the maximum humidity be limited. As was shown last month, it is generally difficult under conditions of light load, and especially in the in-between seasons of spring and fall, to maintain humidities even within a satisfactory limit. Reheating is a necessity if satisfactory year 'round conditions are to be maintained. Reheating is not only desirable because of the fact that it permits temperatures and humidities to be maintained independently of each other in zoned air conditioning systems, but it is also one of the most satisfactory methods for permitting certain zones to be heated while others are cooled, a condition which is frequently necessary in spring and fall.

Although the first cost of installing a reheating system is satisfactory, the operating cost is higher than for any other type of system. The reason is, of course, that regardless of the actual load on the system, the sensible heat added by reheating is always enough to maintain the refrigerating load at the maximum capacity. Consequently the refrigerating equipment is always operating at peak load regardless of how light the actual

**Fig. 3—Reheating for a steep ratio line**



load is. Under some conditions, such operation may be justified.

### Primary Air System

To avoid the high operating costs of a reheating system and at the same time to obtain some control or limitation of humidity independently of the temperature control, a third type of system which (for want of a better name) may be called the *primary air system*, can be used in certain types of installations. The primary air system can not be applied as universally as the reheating system. It has two limitations which must be observed in applying it:

1) It can be used only where the sensible heat load under maximum load conditions is high, say about 90 per cent of the total load. Under light load conditions, the sensible heat load should not be less than, roughly, about 65 per cent of the total load.

2) In zoned systems it can be used only in installations where the character of the loads at any one time is about the same.

The discussion in the following paragraphs will describe the primary air system and at the same time illustrate the reasons for these two limitations.

As stated, the primary air system is limited to installations in which, under maximum load conditions, the sensible heat load is about 90 per cent of the total load. On decreasing sensible heat loads, the primary air system provides control of temperature until the sensible heat percentage has fallen to, roughly, 65 per cent of the total after which the temperature goes out of control unless reheat is provided. However, in many installations—such as offices, hotel guest rooms, and apartments—the sensible heat percentage under maximum load conditions is about 90 per cent and rarely falls through the summer below 65 per cent. For such installations, the primary air system provides a simple and economical method, insofar as operating costs are concerned, for providing independent control of temperature and humidity. The primary air system is not suitable for installations such as theaters, ballrooms, or other places where the sensible heat percentage under maximum load conditions is about 70 per cent because as the sensible heat load falls the sensible percentage falls lower and the temperature goes out of control unless reheat is provided. If reheat is needed for control a greater part of the operating time, then a straight reheat system is, as will be shown later, more economical to operate than a primary air system with reheat.

One elementary form of the primary air system is shown in Fig. 4. Although a zoned system is shown, it is equally applicable to a single fan system. In the primary air system of Fig. 4, two streams of air are conditioned separately. One stream is called the primary air, and the other the secondary or return air stream. The primary air supply is usually either all outdoor air or a mixture of return air and outdoor air. The primary air is generally limited to about 20 or 25 per cent of the total air circulated by the zone fans. The balance of the air circulated by the zone fans is return air. The method of figuring the quantities of primary and outdoor air required will be given later.

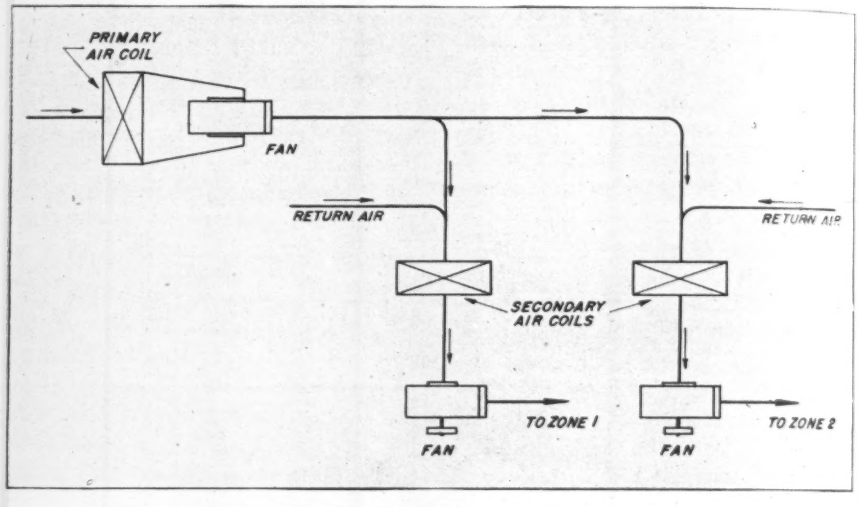



**Fig. 4—One type of simple primary air system**

Only sensible heat is removed from the air by the secondary coils; the temperature of the refrigerant supplied to these coils is maintained at a high enough point so that no condensation of moisture can take place on the secondary coils. The primary air is cooled to a low enough dew point temperature so that all of the moisture load in the conditioned space can be absorbed by the primary air stream alone. The supply of refrigerant to the secondary coils is controlled by the dry bulb thermostats in the return air stream from each zone.

The operation of the primary air system can better be understood by referring to the skeleton psychrometric chart of Fig. 5. Suppose that a room having a ratio line like *1-2* is to be maintained in the condition represented by point *1*. The primary air is always cooled and dehumidified to the low temperature represented by point *8*. Return air from the room in the condition represented by point *1* is mixed with primary air in the condition represented by point *8*. The resulting condition of the mixture is represented by point *9*. The air mixture in the condition represented by point *9* enters the secondary coil and is cooled along a line of constant dew point temperature to point *3* on the ratio line *1-2*.

Suppose that the sensible load in the room falls to such a level that the ratio line is represented by *1-4* instead of *1-2*. In this case the thermostat in the return air would throttle the refrigerant to the secondary coil so that the air mixture, still in the condition represented by point *9*, would be cooled only to point *5* on ratio line *1-4*. As long as the percentage of primary air in the total mixture circulated by the zone fans remains the same, and as long as points *1* and *8* remain in the same location, point *9* will also remain in the same location regard-

less of the changes in the room ratio line.

To continue with Fig. 5, if the sensible load in the room should fall so that the ratio line in the room is now represented by *1-6*, the mixture in state *9* entering the coil would be cooled only to point *7*. Finally, if the sensible load should fall to such a condition that the ratio line was represented by *1-8*, the mixture entering the secondary coil in state *9* would also leave in state *9* because in this case the thermostat would completely cut off the refrigerant to the secondary coil. It is obvious that the lowest sensible load and the lowest ratio line for which complete control of relative humidity can still be maintained is ratio line *1-8*. Any steeper ratio line will result in a decrease in dry bulb temperature in the conditioned space unless the primary air is reheated. It is for this reason that the primary air system is limited in its application only to those types of installations in which the sensible heat percentage under maximum sensible heat load is rather high.

If the maximum sensible heat ratio is low under conditions of maximum load, the range over which the system could be controlled without reheating the primary air would be too limited to be of value. Thus in Fig. 5 if the ratio line under maximum sensible load conditions is only *1-6*, the range of ratio lines in which control could be maintained would be the narrow one from *1-6* to *1-8* with a range in supply temperatures from points *7* to *9*. This frequently may be much too small to be of any practical value, especially in view of the fact that in such systems the ratio line under light loads might frequently swing to the right of *1-8*. If under maximum load conditions the ratio line is so steep as to require reheating, a straight reheating system will be more economical to operate—and to install—than a primary air system. The range of ratio lines in relation to the magnitude of the range in sensible loads will be discussed later.

### Chemical Dehumidification

Without reheating, the primary air system can be freed of the first limitation previously mentioned only by using chemical means, such as silica gel or lithium chloride, for dehumidifying the primary air. Thus in Fig. 5 the primary air can be dehumidified chemically to a point like, say, *11* and then recooled by means of either city water or water from a cooling tower to a point like *12*. Point *11* would lie above the wet bulb line through point *1*. The dry bulb temperature at points *1* and *12* should be approximately the same. In this case, the primary air handles only the moisture load while all of the sensible cooling is handled by the secondary air.

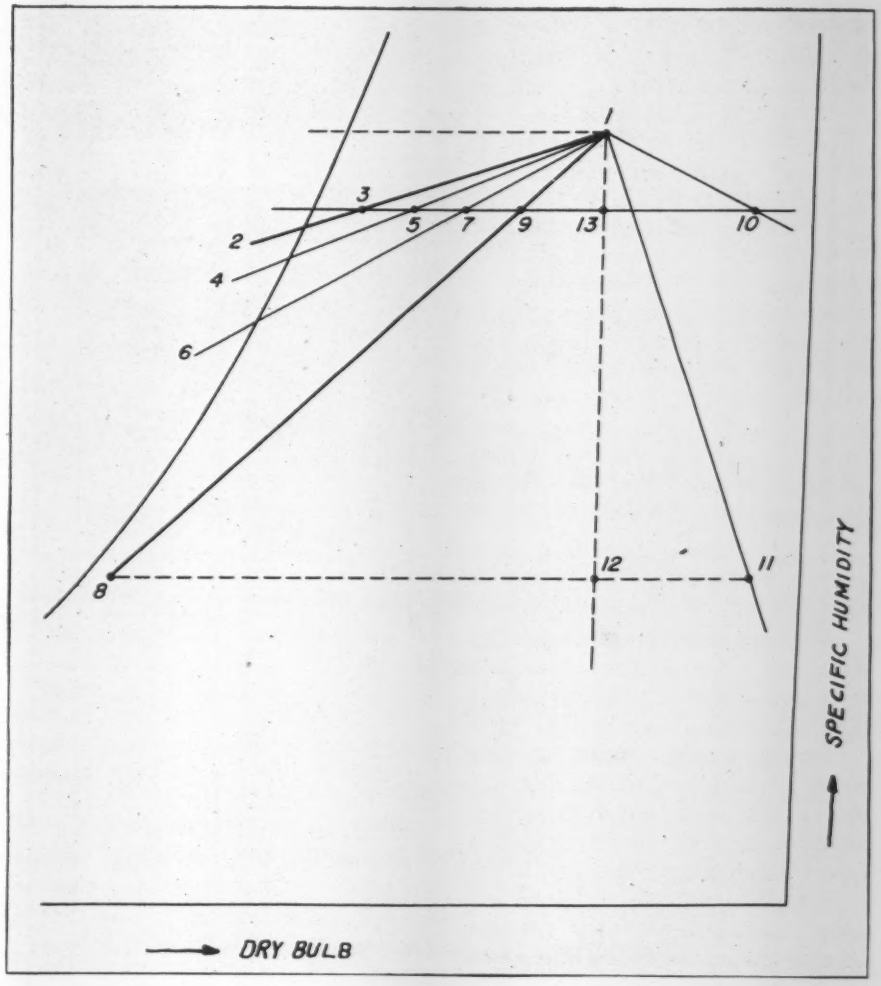



Fig. 5—Primary air cycle on the psych chart

From Fig. 5 it is obvious that the primary air system can now handle any ratio line lying anywhere between lines *1-2* and *1-12*. In other words, regardless of what percentage of the total load is sensible heat, the primary air system with chemical dehumidification is flexible enough to be used under all sensible load conditions. However, separate dehumidifying equipment for the moisture load and separate refrigerating equipment for the sensible load has until now seldom been warranted except in large installations.

The operation of a primary air system during the in-between seasons of summer and fall will be discussed in some detail later. However, in order to explain the second limitation in the application of the primary air system, it is necessary to discuss here, briefly, the functioning of the primary air system for heating during the winter months.

### Operation During Winter

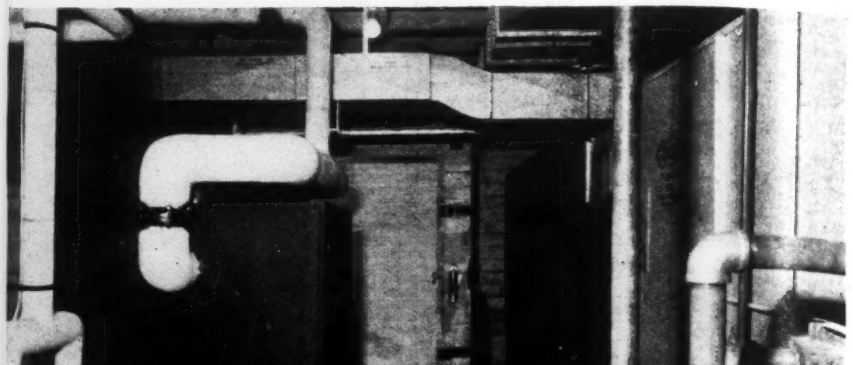
During the winter the circulating water supplied to the secondary coils is heated. Consequently the air mixture flowing through the secondary coil is heated to a high enough temperature to offset the heat losses in the conditioned spaces. For example, referring to Fig. 5, suppose that primary air in the condition represented by point

*8* is mixed with return air in the condition represented by point *1*, resulting in a mixture at point *9*. The air entering the secondary coil in the state represented by point *9* is heated to the state represented by point *10*. The warm air is then supplied to the conditioned space. (Point *8*, representing the state of the primary air, can, of course, be at any location on the psychrometric chart necessary to produce the proper humidity in the conditioned room; it has only been left as shown in Fig. 5 for convenience, since its exact location is immaterial for the purposes of the present discussion.)

During the spring and fall months, if no heat is needed in the room, the thermostat will cut off the supply of warm water to the secondary coil and air can be supplied to the room in the condition represented by point *9*. In this condition a small amount of cooling is possible in the room. Usually the sensible heat load that can be absorbed by the primary air is about 25 per cent of the maximum. If in a number of zones some require heating and some require cooling, the primary air system will be satisfactory as long as, when the system is on the heating cycle, the maximum sensible heat gain is less than roughly 25 per cent of the maximum summer sensible heat gain. However, if, while the system is on the heating cycle, some zones have cooling loads that are almost as great as the summer cooling load, the primary air system will not be satisfactory.

For example, an installation might consist of both offices in the interior of a building and offices along the outside walls of the building. During the heating cycle, the interior offices would probably have a cooling load about the same as in summer, while the offices along the outside wall would require heating. For such an installation, a primary air system would not be satisfactory.

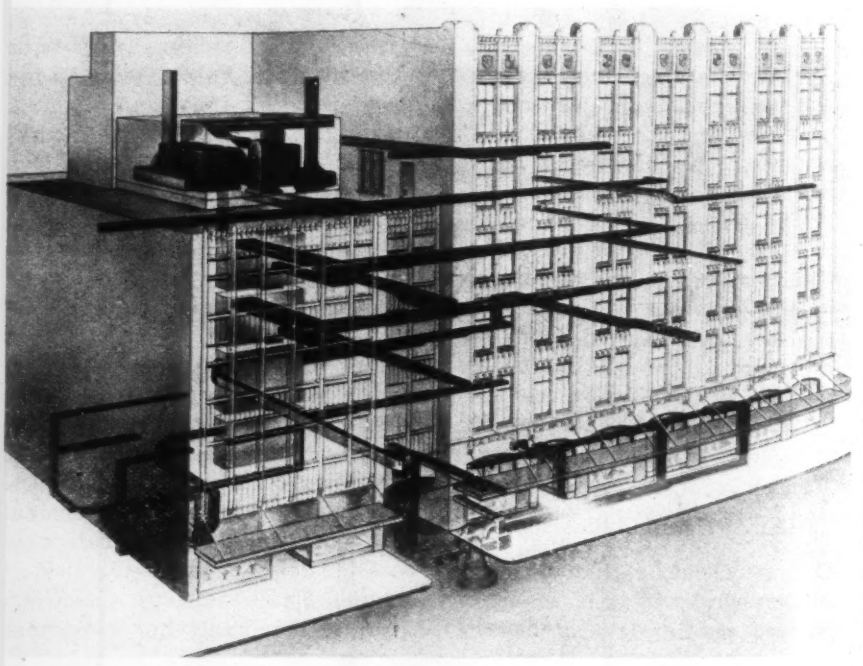
For satisfactory results from a primary air system, the character of all the loads must be approximately the same. Systems in which different zones have large heating and cooling loads at the same time cannot be served by a system of this type. For installations in which both large heating and cooling loads are encountered simultaneously in different zones, the reheating system is the most flexible. If the installation is extensive enough, a reheating system can be provided for the interior spaces and a primary air system for the exterior spaces. There are, of course, many variations in system design that can be used to fit the problems arising in particular installations.



**Silica gel dehumidifying installation at a telephone company, where humidity control is used in order to prevent cross talk and other difficulties in exchanges.**

When applying the primary air system to a zoned installation or, what amounts to the same thing, an installation consisting of many separate rooms, care must be taken to be sure that the character of the loads in the various zones is approximately the same. The primary air system is not suitable for zoned installations in which the character of the loads in the various zones is widely different at the same time.

**Large silica gel dehumidifier installation at a department store. There is a total of 36,000 cfm of dehumidifier capacity handling all outside air on this job. This dry air is directed to individual floor circulators to handle the latent load in these spaces. Cold well water is used for cooling. (Photos courtesy Bryant Heater Co.)**



# Planned Conservation Program Saves Fuel at Large War Plant

**When W. S. Bard wrote this article he was employed in the steam generation and distribution section at a large ordnance plant; now he is learning to use some of the materials he formerly helped to produce. He describes here in detail the various methods investigated at this plant to better match heating steam consumption to the heating load, and shows the results which have been achieved. Because of the care with which he describes the fuel saving program for this particular case . . . and because of the increasing importance of conserving fuel . . . no one concerned with the heating of a large plant or building will want to miss this worthwhile, practical contribution to a timely subject. An interesting angle is the use of a color coding system to show which radiation and steam using equipment should be turned on as the outdoor temperature drops or the occupancy changes.**

CONSERVATION of fuel—be it oil, coal, or gas—is becoming increasingly important every day. The saving of a few gallons of oil or tons of coal no longer represents only so many dollars and cents saved, but a man hour saved, a railroad car saved—effort saved which can be diverted to making and transporting arms and ammunition for our fighting men.*

In order to judge the results achieved by a conservation effort, it is necessary to have a standard against which to compare the amount of fuel used before and after the program is inaugurated. Since fuel consumption varies with the outside temperature, this standard must take into account variations in temperature. Such a standard was described in the articles on How to Watch Fuel Use in the February, March and April issues of HPAC.

By plotting daily fuel consumption during a representative or basic period against temperature deficiency as measured by degree days the dependency of fuel consumption upon the outside temperature is shown. This relationship can be described by a smooth curve, Fig. 1, which represents average or "normal" fuel consumption.

### Steam Conservation Experiment at Large War Plant

The deviations of actual consumption from "normal" consumption during the basic period can be described by an average or "normal" deviation. The actual deviation can be compared to this "normal" deviation in the form of the deviation ratio (the actual deviation divided by the "normal" deviation), a figure which allows unit to unit and day to day comparisons of performance. The distribution and relative variation of these ratios during the basic period may be considered to be representative of the performance to be expected during succeeding periods, and reports for these succeeding periods may be judged on the basis of previous experience.

This system was applied to the fuel consumption data for a large war plant, and led to the development of the theory that the deviations of actual from "normal" consumption are caused primarily by the failure to proportion the amount of radiation turned on to the outside temperature. If a system could be devised which would lead to a better proportioning of effective radiation to outside temperature, the range of the deviations should be reduced and a saving made by the prevention of overheating.

At this plant it is the responsibility of the pipefitters assigned to each line to control the radiation. In the absence of automatic control equipment, it is necessary to depend on these pipefitters to regulate the radiation to maintain the proper building temperatures. A conservation experiment was undertaken in one unit to determine how great a saving, if any, could be made by closer regulation of the radiation. This program attempted to effect the saving in two ways:

1) By attempting to proportion effective radiation to the outside temperature so as to maintain building temperatures at the minimum consistent with the comfort of the occupants and the demands of the manufacturing processes.

2) By the shutdown of ventilating equipment and the lowering of building temperatures during periods of nonoccupancy.

Room thermometers were installed throughout the buildings in the unit so that building temperatures could be watched.

In order to aid the pipefitters in proportioning the radiation turned on to the outside temperature, the radiation and steam consuming equipment in the buildings was divided into the following groups and the members of each group identified by a spot of color painted near the valve.

### Radiation and Steam Using Equipment Color Coded

1) *Orange*—To be shut down during periods of nonoccupancy. This color was assigned to the ventilating equipment and indirect heaters which could be shut down

*Of which the author is now one.—Ed.

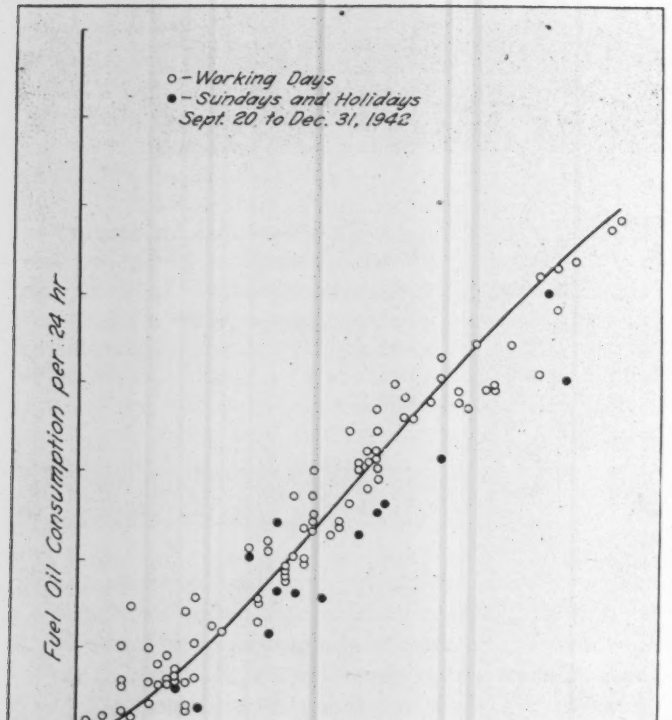

o - Working Days
• - Sundays and Holidays
Sept. 20 to Dec. 31, 1942
Fuel Oil Consumption per 24 hr
FIG. 1
0 5 10 15 20 25 30 35 40 45 50 55 60
Degree Days

**Fig. 1—Daily fuel consumption plotted as a function of daily degree days. Note that the relationship can be described by a smooth curve drawn in such manner that about one-half of the observations lie on either side. This curve represents "normal" daily fuel consumption and is used in comparisons**

while the buildings were not in production.

2) *Brown*—Generally shut off, open only on request of the operating department. This color was assigned to radiation which experience had shown to be unnecessary to heat the buildings even in cold weather and to radiation in certain unoccupied rooms.

3) *Unmarked* — Operating continuously, close only on request of the operating department. Most of the processing equipment, some of the automatic heating equipment, and radiation in rooms in which high temperatures are maintained because of the manufacturing process were left unmarked.

4) *Black*—Regulate by hand according to the demands of the occupants. This color was assigned to the radiation in small rooms such as offices in which there were only one or two radiators which must be regulated by the radiator valves.

The bulk of the radiation was divided into five groups to be operated in accordance with an outside temperature schedule:

5) *White* — Open when the outside temperature is below 60 deg.

6) *Green* — Open when the outside temperature is below 45 deg.

7) *Blue* — Open when the outside temperature is below 30 deg.

8) *Yellow* — Open when the outside temperature is below 15 deg.

9) *Red* — Open when the outside temperature is below 0 deg.

Each of these last five groups contains approximately one-fifth of the radiation in the manufacturing sections of the buildings. The steam condensed in the piping suffices to heat the buildings when the outside temperature is above 60 deg.

### Weather Predicted for Coming 24 Hr Period

Weather predictions for 24 hr in advance were obtained each afternoon from the plant weather station. Instructions based on these predictions were telephoned to the pipefitters in the unit advising them when to turn on or off the appropriate groups of radiation in accordance with the outside temperature schedule. They were instructed also to watch the thermometers in the buildings and to turn the radiation on or off in a particular group as the temperature in the building rose above or fell below 70 deg. Thus if the outside temperature were 2[illegible] deg. it should be necessary to have the radiation in groups white and green turned on completely, but only a portion of the radiation in the blue group.

An outside temperature thermometer was installed near the unit so that the pipefitters might know the actual outside temperature. Knowing this, and the temperature to be expected during the next few hours, they could anticipate the fluctuation in load and thus change the amount of radiation turned on in advance of the change in load. The color coding provides a reasonably accurate method of gaging the relative amounts of radiation turned on or off. The success of the program depends on these men understanding the system and their willingness to cooperate.

The daily and monthly fuel consumption figures for this unit when compared with the figures for the other units by the methods outlined in the February, March, and April issues showed an immediate drop. During the first month fuel consumption was 5 per cent lower than the estimated consumption had not the program been in effect. During the second month these savings increased to 8 per cent. (Fig. 2.)

### Bi-Hourly Fuel Consumption

Previous to the conservation experiment, all correlations between fuel consumption and outside temperature had been based on fuel consumption over a 24 hr or longer period. However, the oil meters in the various boiler houses were read every hour and the official thermometers at the plant weather station were read every 2 hr. It was decided to compare actual bi-hourly fuel consumption with "normal" bi-hourly consumption. "Normal" bi-hourly fuel consumption was determined for a given period by taking one-twelfth of the "normal" daily fuel consumption for a corresponding temperature deficiency below 65 deg. Thus if the outside temperature at 2:00 a.m. was 46 deg. the "normal" fuel consumption from 1:00 a.m. to 3:00 a.m. was one-twelfth of the "normal" daily oil consumption for 19 degree days, as determined from the "normal" curve for the unit under con-

sideration. This involves the assumption that the non-heating load —processing uses, hot water heating, and line losses—remains fairly constant throughout the 24 hr period, an assumption which proved to be nearly correct at this plant.

Actual and "normal" bi-hourly fuel consumptions for three units are plotted in Fig. 3. Diagram *A* in Fig. 3 is the graph for one of the units in which about 75 per cent of the heating is done by thermostatically controlled indirect heating systems, one of few such units at this plant. Note that the curves remain very close together throughout the day.

Diagrams *B* and *C* are for units with hand controlled standing radiation. Diagram *B* is typical of these units on a day that is warmer than its predecessor; the curve of actual consumption cuts across the valley of the curve for "normal" consumption, since radiation turned on during the colder night hours is not turned off during the day when the outside temperature is warmer, and actual consumption for the day is above "normal." On a day which is colder than its predecessor consumption would be below "normal" because the curve for actual consumption would tend to cut across the peak of the curve for "normal" consumption.

Diagram *C* is for the unit in which the conservation experiment was undertaken. The effect of the conservation effort upon the curves was that the curve for actual consumption cut across the peaks and more closely followed the valleys of the curve for "normal" consumption. This particular unit was in operation 24 hr per day at the time, indicating that the "normal" allowances are liberal, since the building was adequately heated during the night hours, yet consumption was considerably below "normal."

Inasmuch as the automatically controlled unit (diagram *A*) was not in operation during the midnight to 8:00 a.m. shift at the time, the curves suggested the possibility of savings if the indirect heating units were partially cut out of service while the buildings were not occupied. Instructions to shut off one-half the units entirely and lower the thermostats for the remaining half by 10 deg during this period were sent to the unit, with the result that fuel consumption during this period dropped considerably below "normal."

A similar schedule of lowering building temperatures during periods of nonoccupancy was also put into effect on the experimental line.

The high correlation between temperature and fuel consumption over such a short period of time in the predominantly automatically controlled unit emphasizes the dependency of consumption upon outside temperature when the heat supply is regulated by the actual load. This suggested that it would be well to attempt to regulate effective radiation even more closely by the outside temperature than had been so far attempted in the other units.

### The Fuel Conservation Plan

The investigation of bi-hourly fuel consumption indicated that the attempt to proportion radiation to temperature and to cut down building temperatures in periods of nonoccupancy was definitely a step in the direction of fuel saving, but that an even closer control than the color coding system was desirable. In order to extend the conservation effort over the entire plant as quickly as possible and at the same time provide this closer control the following plan was formulated and adopted in all units:

1) Clerks were assigned to cover a telephone desk in the office of the chief boiler house engineer. The coverage is for 24 hr per day seven days per week.

2) At the end of each hour the boiler house engineers report over the telephone to this desk their oil and feedwater meter readings and other pertinent data.

3) The plant weather station was requested to report official temperature readings to this desk each hour (a suitable outside thermometer installed where it might be read by the clerks would have been satisfactory had not the weather station records been available). The weather station also reports the current weather predictions to this desk.

4) At the end of each hour the clerk on duty computes the amount of oil which has been burned in each boiler house during the preceding hour and compares this with the amount which should have been burned during this period according to the outside temperature. Tables showing the amount which should be burned per hour in each boiler house for various average temperatures are furnished to the clerk. These amounts are computed by dividing the "normal" daily oil consumption by 24 for the corresponding temperature deficiencies below 65 deg. These tables may be revised from time to time as conditions in the various units change. Average hourly temperature is computed by averaging the outside temperature at the beginning and end of each hourly period. All rec-

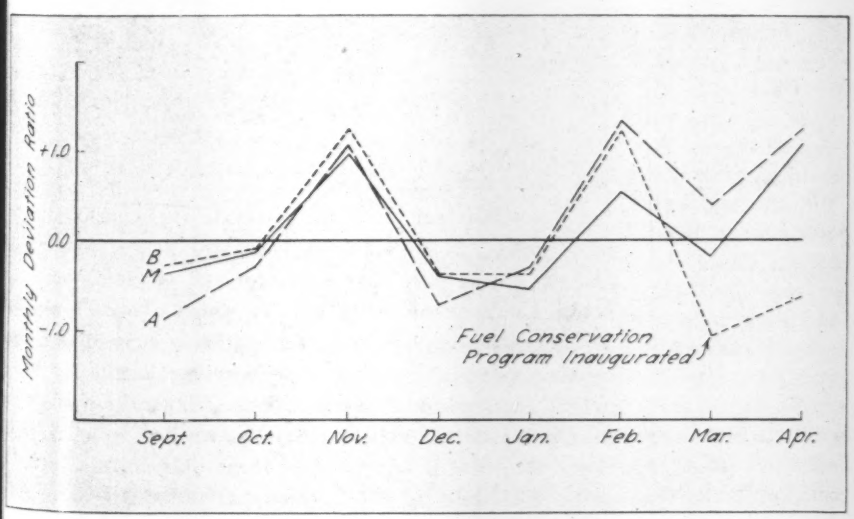



**Fig. 2—Comparison of the monthly deviation ratios for two typical units (curves *A* and *B*) with the median monthly deviation ratio for all units (curve *M*). Note that through the month of February the curves for units *A* and *B* lie very close to each other and to the median or average curve for all units combined. During March the conservation program was started in unit *B*. Results are obvious**

ords are kept on a standard report form (Fig. 4). If the amount of oil which has been burned during an hourly period is significantly more than "normal" in any unit, the clerk notifies the pipefitters in that unit to cut down the load by shutting off part of the radiation in the unit.

5) At the end of each hour the clerk on duty computes the efficiency and percentage of rating at which the boilers in each boiler house have been operated during the past hour. This is determined by reference to graphs, showing the efficiency and rating for the amount of oil consumed and the amount of feedwater evaporated. If the efficiency is low he will notify the boiler house engineer. If the boiler house engineer cannot correct the condition in the next few hours, the clerk notifies the oil burner maintenance department.

6) On the basis of the amount of fuel burned and the feedwater consumed during the past few hours, and the anticipated consumption during the next few hours based on the weather predictions, the clerk in conjunction with the boiler house engineers determines how many boilers shall be in operation in each boiler house and when boilers shall be added to or removed from the line. This is to insure that the boilers will be operated at as high a percentage of rating as is practical.

7) The clerks keep all records pertaining to boiler house operation, personnel, etc., and prepare all reports, so that their time while waiting for the hourly reports is utilized.

As this program was adopted in the various units an immediate drop in fuel consumption below "normal" was noticeable.

This program was put into effect in these units at such a time that the heating season was over before a full month's records were obtained. Daily and weekly comparisons showed that the savings in most of these units were even greater than the 8 per cent anticipated from experience in the experimental unit.

### The Summer Conservation Program

The conservation effort was continued on into the summer months, although the fuel used to supply steam for the non-heating uses—hot water, processing, and line losses—was only a fraction of that used during the heating months. Hourly records of fuel consumption were kept and telephone contact with the line pipefitters was maintained. Hourly "normals" for the non-heating uses were set up for each unit by dividing the average daily consumption during the preceding summer by 24. While checking continuously on hourly consumption, an effort was made to cut down this consumption by first cutting off all low pressure heating mains as soon as the weather allowed, and later by cutting off steam in the high pressure lines leading to buildings that did not need steam in the summer. When the weather got colder for a short period these mains were turned back on, but turned off again as soon as possible. Later the steam supply in all high pressure lines was shut off, and the boilers shut down, during extended periods of nonoccupancy such as weekends. Fuel consumption for the area for the month of July 1943 was 21 per cent less than during July 1942. The bulk of this saving was probably the result of the conservation effort, although accurate comparisons are difficult because of changes in the level of activity in the various units from one

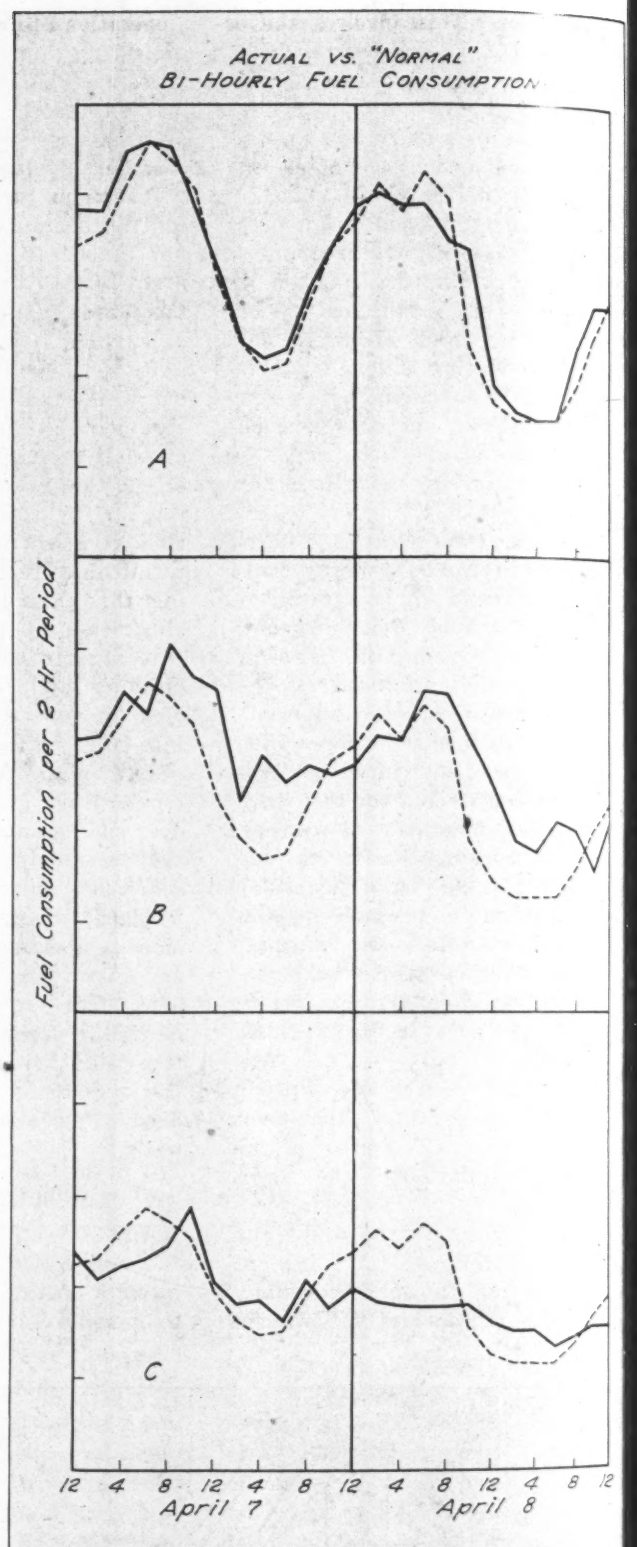



**Fig. 3—Actual and "normal" bi-hourly fuel consumption for three units. The solid lines represent actual consumption and the dotted lines represent "normal" consumption. Diagram *A* is a graph for a unit in which about 75 per cent of the heating is done by thermostatically controlled indirect heating units. Note how close together lie the curves for actual and "normal" consumption. Diagrams *B* and *C* are graphs for units heated by hand controlled standing radiation. Diagram *B* is typical of these units while diagram *C* is for the unit in which conservation efforts were made**

DAILY LOG

Boiler House _____ Date _____

| Time | Boiler # Meter Reading | Boiler # Oil Use | Boiler # Meter Reading | Boiler # Oil Use | Boiler # Meter Reading | Boiler # Oil Use | Total Hourly Use | "Normal" Hourly Use | Over or Under | Outside Temp. | Feedwater Meter Reading | Feedwater Use | Ave. Eff. | Ave. Rate | Remarks |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 12:00 | | | | | | | | | | | | | | | |
| 11:00 | | | | | | | | | | | | | | | |
| 10:00 | | | | | | | | | | | | | | | |
| 9:00 | | | | | | | | | | | | | | | |
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**Fig. 4—Daily record form for hourly fuel consumption. The meter readings are entered from the bottom up for ease of subtraction. One of these sheets is kept each day for each boiler house**

year to the next. During the heating months these changes in activity had little effect on consumption, since so long as the buildings had any occupants at all they were heated and only the quantity of steam used for processing changed with changes in activity. This represented only a small proportion of the total consumption. During the non-heating season the processing uses represent a high percentage of total consumption.

### Final Recommendations on Fuel Saving Methods

The general conclusion drawn after studying the results of the various conservation efforts was that both the color coding system and the hourly comparison of actual with "normal" consumption aided in the proportioning of effective radiation to temperature, with the resultant saving of fuel, and that the hourly check with its closer control was the more desirable. The results achieved by the summer conservation program also were surprisingly good. The recommendation for the following heating season was that the hourly check system be retained, but that it be combined with the color coding system, since the color codes provided a reasonably accurate method of gaging the relative amounts of radiation turned on during any given period.

## Measuring the Flow of Fluids

THE EDITOR—

I found William Goodman's series of articles on Measuring the Flow of Fluids, published in the August 1943 - February 1944 issues of HPAC, very instructive on account of the concise and clear way in which the subject matter is presented. For example, the installment on the calculation and choice of an orifice as presented there is much easier to grasp and more direct for everyone than is the formal presentation of the ASME. The thing has been cleared of every ambiguity by Mr. Goodman and this is especially helpful for one who is not a specialist in these metering problems. In the same way the presentation of flow coefficients against a modified Reynolds number (which appealed to everyone, probably) tends to make more usable and easier a study of this question. Some other parts of the paper contain in a renewed form material which had previously appeared in another of Mr. Goodman's series. (How to Protect Equipment Against Failure of Flow, HPAC, June and July 1942).

I wish to state that I am very pleased with the character of your publication, which I have been reading for the last six years.—LEONARD CARTIER, C. E., department of hydraulics, Ecole Polytechnique, University of Montreal.

# "OPEN for DISCUSSION"

**HPAC's readers and editors will welcome your comment on articles or parts of articles published in these pages, and on other topics of interest to those concerned with design, installation, operation, and maintenance of heating, piping, and air conditioning systems in industrial plants, commercial, institutional, and public buildings . . . Address The Editor, Heating, Piping & Air Conditioning, 6 North Michigan Avenue, Chicago 2, Illinois.**

## Three Reasons for Making Your Postwar Plans Now

THE EDITOR—

There are three good reasons why every concern should make as complete construction and modernization plans as possible now for the postwar era:

1) During the war period, due to scarcity of materials and shortage of mechanics, much replacement and maintenance has been postponed. There is a limit to which this postponement can be carried if equipment is to be kept in operation, and wise organizations will survey their equipment and be prepared to move as soon as conditions will permit the work to be started.

2) During the war there has been virtually no replacement due to obsolescence and an accumulation of jobs should exist.

3) Many plants will be faced with peacetime problems that will be entirely different from their wartime operation and all far sighted managements will have their plans ready for conversion to peace operation as soon as the demand due to the war is over.

In a great many instances piping supplies, heating equipment, and air conditioning devices will be scarce and these things cannot be created overnight. The concerns who have their plans ready and immediately go into the market and place their orders will undoubtedly be very much earlier served than those who do no planning and who have not prepared for the solution of their problems.—WILLIAM A. HANLEY, Eli Lilly and Co. Member of HPAC's board of consulting and contributing editors.

## Now Is the Time to Adopt Metric System

THE EDITOR—

I read with exceptional interest the short comment under the above title on p. 211 of the April HPAC because I have done some agitating on the subject myself and am heartily in favor of making the shift as soon as possible.

But here is a statement by our National Bureau of Standards in which that bureau says, "It is generally agreed by those favoring adoption of the metric system in this country and by those who oppose it that this is not a proper time to devote any appreciable time or energy to arguing the merits of this question or carrying on extensive propaganda in favor of either side of the question."

The bureau is neutral. It says, "Those desiring arguments in favor of its compulsory adoption are referred to the Metric Association, Pottsville, Pa. Those desiring arguments against it are referred to the American Institute of Weights and Measures, 33 Rector St., New York City."

It might not be out of place for this writer to state that practically every country of major importance has already accepted the metric system—all but the United States and Great Britain.

Also, the writer finds that but few people are aware of the fact that the famous James Watt, an English engineer and scientist, was the originator of the decimal system of measurement. I had always considered it of French or German origin myself, but that is not the case. Nor is it well known that our present "English system" is of German origin. We, therefore, have this interesting situation—the English speaking people are clinging to a German invention, which the Germans themselves have discarded, and the balance of the world is using an English invention. That should be a good one for "Believe-It-or-Not" Ripley.

### Approximate Relation

The simplest way I know as an aid toward remembering the approximate relation between the units in the two systems is the following:

Length: One yard plus 10 per cent equals one meter.

Weight: Two pounds plus 10 per cent equals one kilogram.

Volume: One quart plus 5 per cent equals one liter.

Concentrate on the above easily remembered table for a few minutes. You add 10 per cent in each case excepting "liquid measure," and there you add 5 per cent. Associate "liquid" with the old pre-prohibition beer at 5c per glass and you'll never forget it.—W. F. SCHAPHORST.

● *It is a patriotic duty to save fuel by heating efficiently.*

# SAM LEWIS' PAGE

**[In this regular feature, informal comment on heating, piping and air conditioning problems and their solutions is given — at the request of the editors — by Samuel R. Lewis, consulting engineer, and a member of HPAC's board of consulting and contributing editors]**

## Central Hot Water Heating

SEVERAL inquiries have recently been made concerning central forced circulation hot water heating, in which the water is pumped directly through the boilers.

Failure by the industry in general to take advantage of the superior heat transfer facilities afforded by this shortcut method has occasioned some surprise. I know of a very large electric manufacturing concern which has employed this method with entire success for many years. A steel foundry has been heated for 10 years by radial flow downward discharge, unit heaters and ventilating machines using water which is pumped through the boilers and which is not allowed to make steam. The water flows rapidly around the heating surfaces in the boilers, which are fitted with special interior deflecting baffles to insure the best possible contact. There is no mud or scale in the boilers, and no elaborate water treating equipment is necessary—though the water supply is loaded with precipitable salts.

There are many institutions and housing projects in which forced hot water is used as the heat carrier. The usual practice is to install the pump in the return main near the boiler. An excellent means of temperature control is to install near the pump a three way valve or its equivalent in separate valves, arranged and controlled so that when the building is sufficiently warm, most (if not all) of the circulating water will bypass the boiler and thus will cool gradually. If heat is needed, some of the hot water from the boiler will be mixed with the recirculated water.

When there are several buildings served by such a system, it is practicable to achieve low cost automatic temperature control for each building by installing a separate pump and mixing valve in each building. Since the acceptable location for a single thermostat to control the temperature in a large building usually is difficult to find, the thermostat may be placed outdoors so that as the outdoor temperature lowers, the water temperature increases. These schemes operate with general satisfaction.

The pump in the boiler house merely insures that the water in the mains shall be ready to serve the auxiliary pumps in the individual buildings, generally should operate continuously, and may in turn recirculate part of the water and bypass the boiler, governed by an outdoor thermostat. The temperature of the water in the boiler may, via an aquastat, govern the fuel consumption.

The piping systems in the individual buildings may be single pipe or double pipe, depending on physical conditions, size, or designer's whim—since either type of piping will operate satisfactorily if intelligently designed. The trunk mains serving the various buildings generally are provided with direct returns (not first flow-last return), since adjustment of flow to the various buildings by valves is not difficult. If the individual buildings of a large central system each have thermostatic mixing valves and pumps, the distribution of water to them from the trunk mains is rarely troublesome. When the first building to get warm commences to be throttled, the unused water tends to accelerate the service to the remaining buildings.

There are many very extensive central hot water heating systems for buildings of various ages and uses. Old central plants, such as for one of the state universities, have heating systems in individual buildings of widely different resistance to water flow. In such cases the quickest and most satisfactory solution is to install for each building a local mixing valve and pump.

It must be remembered that with single pipe hot water systems of large size the water temperature will be reduced as successive heat transmitters along the main extract the heat, so that the heat transmitting surface should be progressively increased along the flow main to compensate. This somewhat tedious computing of the necessary increase in size for successive radiators is not customarily undertaken for small buildings.

● *A program of installing stokers and otherwise modernizing its heating plants has been inaugurated by the Birmingham, Ala., city board of education, which operates 70 high schools and elementary schools including 100 buildings. The program ties in with the city's smoke abatement campaign and is designed also to save fuel and provide more even temperatures for the schools. Installations have already been made in three of the schools and others will be made each year from allotted funds, first in the high schools and then in the elementary schools, according to the Stoker Manufacturers Association. "We are convinced that the saving in fuel will be enough to liquidate the cost of the equipment," said R. C. Cornic of the board of education.*

**Eleven Story Office Building Modernizes, Gets 6½ Per Cent Saving, Better Heating**
**By . . . . . P. A. Hyde**

**The war effort depends upon efficiency in manufacturing and business operations; promoting such efficiency, too, is one of the most practical methods of preparing now for the postwar period of rehabilitation and competition. Mr. Hyde, superintendent of steam sales, Akron division, Ohio Edison Co., describes here the modernization of the heating system in a typical office building, and shows the greater efficiency—and dollars and cents savings.**

# HEATING SYSTEM CONTROL PAYS DIVIDENDS IN WAR AND PEACE

EFFICIENCY in manufacturing and business operations is essential to the war effort and, according to many authorities, is an important postwar goal as well. Material and manpower shortages now mean that efficiency earns an "extra dividend"; in the postwar period of rehabilitation and competition, this will be equally true.

Modernization of plant and building heating systems and the installation of temperature control to conserve fuel or steam is thus a contribution to the war effort, as well as a most practical form of "postwar planning." The savings that may be made in this way are most attractive. For example, steam savings effected by 36 installations of temperature control in customer's premises averaged 14 per cent, and the number of years required to pay off the investment averaged 4.7, according to a report given before the National District Heating Association last June.

The purpose of this article is to report the results of a steam saving program inaugurated some years ago in an 11 story office building, equipped with a one pipe steam system. The building, which is exposed on all four sides, has a volume of 1,805,229 cu ft. There is 24,000 sq ft of installed radiation.

At the beginning of the 1934-35 heating season, we contracted with the owners of this building for all of their heating requirements. The method of control at the time was a roof thermostat operating the regular off and on motor operated valve. Radiators were equipped with carbon post vents to an air line. Air removal was by a belt driven pump which, at its best, was rather poor. It was necessary to carry from 9 to 10 psi pressure on this system.

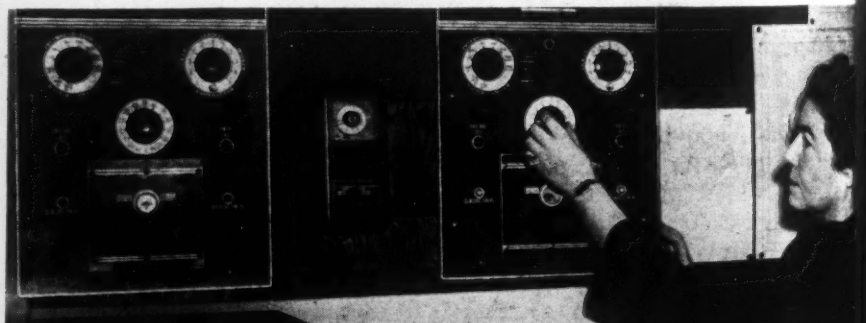
### Vacuum Pump Pays for Itself in One Season

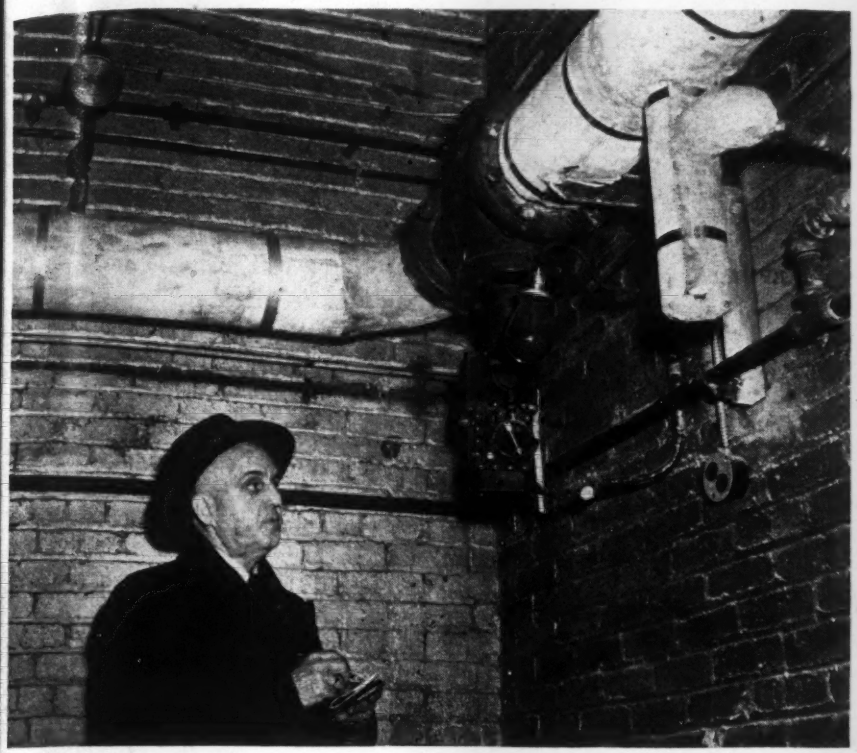
After a few years, we suggested to the building owners that they purchase and install a 40,000 sq ft air line vacuum pump. We guaranteed the pump would pay for itself in steam savings and better operating conditions in three heating seasons; it paid for itself the first season. The following letter from the building manager speaks for itself:

> The board of directors has instructed me to express its appreciation for the manner in which this installation was handled and for the savings it has made in our heating costs. We know you will be interested to learn that sufficient savings were made in the first year to practically cover the cost of this installation. The board also wants to take this opportunity of thanking your representative for his excellent cooperation in assisting us to lower our steam costs.

Some interesting problems were presented when the vacuum pump was put in operation, especially when steam was off at night. We

**Control panels, showing time switch for night shutoff. The operating schedules for the zone system may be seen at the right of the building manager's secretary**





**Modulating valve on one zone of the heating system, 20 per cent open at the time the picture was taken. Vacuum gage shows 15 in., with 20 F outside**

found that the pump running all night created a high vacuum in the end of main drips and held the condensate there until the steam broke the vacuum. We wired a mercury switch to the motor operated valve so that the pump shut off about a minute before the steam valve closed and did not start until the steam was on about 2 min. This corrected the trouble.

### Installation of Two Zone Differential Control

In the summer of 1938 the building owners decided to go "all out" for a modern control system and to zone the building two ways. Frankly, I was skeptical about this type of control on a single pipe system.

Two hole orifice plates were installed in every radiator. Traps were installed on all drip points. The temperature control engineers found out, as we predicted, that they could not zone a one pipe system without trapping the ends of all steam mains. This work was done by the building owners, under the utility engineer's supervision. Practically all traps were float and thermostatic traps, with a few exceptions where drips were of small size and radiator traps were used. Since the traps were installed the system has operated with complete success.

### Steam Savings Summarized

The steam consumption figures given in Table 1 were assembled by the steam heating department of our company. The building management reads the meters daily and charts the consumption on the degree day basis.

The utility has no investment in the control. The original control installed by us was returned to us when the new system was put in.

All condensate is returned to our system and our pump is under the heating system vacuum. Steam for water heating is metered separately from the building heating system and is returned through an economizer, which is also under vacuum.

Regarding the control which went into operation during the 1938-1939 season, it will be noted in the figures that follow that there was a drop of some 10 per cent in pounds of steam per sq ft of radiation per degree day. At the start of the 1939-1940 heating season, the system was checked and proper connections made where necessary.

In order to determine the heating steam savings, let us average the pounds of heating steam per square foot of radiation per degree day for the four seasons before modernization (1935-36, 1936-37, 1937-38, and 1938-39) and compare the result with the average for the four seasons after modernization (1939-40, 1940-41, 1941-42, and 1942-43). The "before modernization" figure is the average of 0.0642, 0.0657, 0.0659, and 0.0597, or 0.2555, which is an average per season of 0.0639. The "after modernization" figure is the total of 0.0559, 0.0573, 0.0627, and 0.0631, or 0.2390, which is an average per season of 0.0598. Thus, the percentage saving is 0.0639 minus 0.0598 divided by 0.0639 and multiplied by 100, or 6½ per cent.

**Recording gages in the boiler room, one on each zone and one on the water heater. The indicating vacuum gage (upper left) is on the return line**

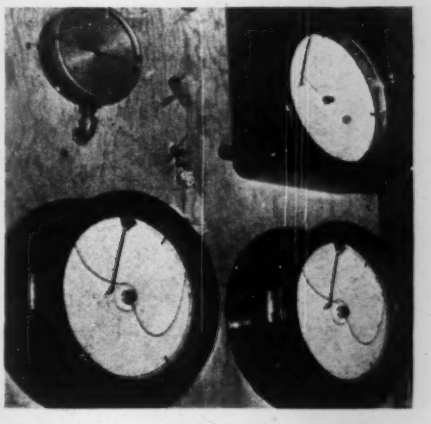


**Table 1—Summary of steam consumption**

| Heating Season | M Lb Steam, Total | M Lb Steam, Water Heating | Net M Lb, Space Heating | Degree Days | Lb Heating Steam per Sq Ft Radiation per Degree Day |
|---|---|---|---|---|---|
| 1934-1935 | 8908 | 857 | 8051 | 5638 | 0.0595 |
| 1935-1936 | 10517 | 911 | 9606 | 6234 | 0.0642 |
| 1936-1937 | 10037 | 1100 | 8937 | 5670 | 0.0657 |
| 1937-1938 | 9997 | 1146 | 8851 | 5602 | 0.0659 |
| 1938-1939 | 8826 | 960 | 7866 | 5490 | 0.0597 |
| 1939-1940 | 9956 | 1233 | 8723 | 6502 | 0.0559 |
| 1940-1941 | 8906 | 1148 | 7758 | 5640 | 0.0573 |
| 1941-1942 | 8833 | 1045 | 7788 | 5175 | 0.0627 |
| 1942-1943 | 9968 | 888 | 9080 | 5996 | 0.0631 |

**The 11 story building has a volume of 1,805,229 cu ft, 24,000 sq ft of installed radiation, is exposed all sides**

This saving is estimated to amount, in dollars and cents, to something over $300 per season. There are other factors than the dollars and cents saving which must be considered, such as more even temperatures throughout the entire building, few opened windows, better satisfied tenants, etc.

The increase in the last two heating seasons shown in Table 1, 1941-1942 and 1942-1943, is due to an increase in building occupancy from approximately 78 per cent in 1940 to 95 per cent in 1941. Another contributing factor is that due to the war, many of the tenants, chiefly doctors and dentists, have found it necessary to maintain longer office hours than previously. As a result, heating time has had to be extended from 7:00 p. m. to 10:00 p. m. The building management is well satisfied with this explanation of their increased heating costs for these two seasons.

## Names for Things

THE EDITOR—

Referring to Sam Lewis' Page in the February HPAC, I object to "gilled" in preference to finned. The timing of the suggestion is belated, to say the least.

You invited a few equipment name suggestions, so here are some:

1) *Radiator*—All cast iron radiators.

2) *Convector*—Any and all types of encased and finned elements. (We can go along with this one very nicely.)

3) *Pipe coil*—Wherever plain pipe is used as direct radiation, connected with headers, return bends, etc. This is the *only* time the word "coil" should be used as descriptive of building heating elements or for cooling elements.

4) *Unit heater—room type.* All designs. Add the words "down blast" to ceiling type with propeller fan (for differentiation).

5) *Heater unit*—Any heating element in the air stream, finned or otherwise and whether used for indirect heating or for tempering air for ventilation.

6) *Steam boiler*—For steam heating systems only.

7) *Hot water heater*—For hot water heating systems only.

8) *Hot water tank*—For domestic hot water tank.

9) *Heater*—For domestic hot water tank heater.

10) *Cooling unit*—Surface cooler for chilled water, well water, etc.

11) *Unit evaporator*—Room type direct expansion unit cooler.

12) *Evaporator*—For direct expansion unit in air stream.

13) *Dehumidifier*—Instead of air washer when used for cooling.

14) *Refrigerating unit*—For all units which combine compressor and condenser in an integral assembly.

My earliest heating experience was with the Gurney Heater Mfg. Co. The firm name in the early 1890's was Gurney Hot Water Heater Mfg. Co. About 1915 the word "boiler" was substituted for "heater" in connection with the hot water heaters. The change was made at that time by nearly all manufacturers, although I never heard any explanation for it.

Most of the terminology of heating and cooling elements as listed above is the practice of our organization (one of the largest engineering firms in the east). We use it on all types of buildings throughout the country and there seems to be no confusion.—HENRY A. GREEN, JR.

## Corrosion of Pipe In Radiant Heating

THE EDITOR—

There has been some discussion of late concerning the corrosion of pipe used in radiant heating installations buried in concrete floors. Reference is made to p. 137 of the March HPAC.

If care is taken to see that the pipe is completely surrounded by well proportioned and well mixed concrete there would be very little danger of corrosion, because the concrete will protect the pipe.—R. C. FARIS, engineer, The Equitable Powder Mfg. Co.

## Radiant Heat and Cooling

THE EDITOR—

The article on radiant heat by H. M. Hart on p. 73 of the February HPAC is a good, commonsense approach to the radiant heating problem. An added advantage of this system is its possible use for cooling in summer. A complete system of air conditioning should include means for circulating cleaned, fresh air. In winter moisture can be added, and in summer the air can be dried. Chemical dryers are available and may prove less cumbersome than other methods, especially for small installations. With dry air there will be no trouble from moisture on the ceiling or other cooling surface.

Radiant methods of cooling and heating minimize the shock to the persons entering, as they avoid drafts.

It is my feeling that the use of wrought iron pipes will materially reduce trouble from corrosion. When placed in the first floor ceiling they assist in heating the second floor area.

I am glad that the HPACCNA is preparing engineering data on radiant heating. I hope that they will send me a copy of it.—H. D. JAMES, consulting engineer.

● *HPAC's readers and editors will welcome your comment on articles appearing in these pages, and on other subjects of timely interest to heating, piping, and air conditioning engineers and contractors. Address The Editor, Heating, Piping & Air Conditioning, 6 N. Michigan Ave., Chicago 2, Ill.*

# Foundrymen's Group Approves Code Recommending Ventilation Practice

A CODE of recommended practices for industrial housekeeping and sanitation has been developed by the industrial hygiene codes committee of the American Foundrymen's Association and approved by its board of directors as recommended practices for the foundry industry. Section V of the code is devoted to ventilation.

According to the code, "All work rooms in which employees regularly work, other than rooms used primarily for storage and warehouse purposes, shall have not less than 2000 cu ft of air space per person regularly employed, based on gross cubical contents, provided the total projected area of doors and windows opening to the out-of-doors is not less than 12½ per cent of the gross area of the work room; otherwise a system of mechanical air supply shall be provided.

"Where there is less than 2000 cu ft of air space per person regularly employed in a work room, or where the total projected area of all doors and windows opening to the out-of-doors is less than 12½ per cent of the gross floor area of the work room, there shall be mechanically supplied an amount of clean tempered air on the basis of one of the two following rules, whichever gives the greatest amount of air supply:

"1) For every 100 cu ft of air space or fraction thereof, where the air space per person regularly employed in the work room is less than 2000 cu ft, there shall be supplied 2 cfm of air per person, or—

"2) When the projected area of all doors and window openings to the out-of-doors is less than 12½ per cent of the gross floor area of the work room, there shall be supplied 2.4 cfm of air per person, for each 1 per cent or fraction thereof, of the projected door and window openings are less than 12½ per cent.

"All offices, rest, toilet, locker, and lunch rooms shall be provided with natural or mechanical ventilation to maintain healthful conditions during periods of occupancy, as per the following requirements:

"If offices, rest, locker and lunch rooms have windows or ventilating openings opening to the out-of-doors, and the net open area of such windows and ventilating openings is not less than 5 per cent of the gross floor area, no mechanical ventilation shall be required.

"If offices, rest and lunch rooms have less than 5 per cent and more than 2½ per cent of the gross floor area represented by net window and ventilating openings opening to the out-of-doors, there shall be provided a mechanical supply of clean tempered air of not less than 0.6 cfm of air per sq ft of gross floor area. However, if the net open area of the windows and ventilating openings opening to the out-of-doors is less than 2½ per cent of the gross floor area, there shall be mechanically supplied 0.6 cfm of clean tempered air per sq ft of gross floor area and there shall be mechanically exhausted 0.3 cfm of air per sq ft of gross floor area.

"If offices, rest and lunch rooms are located in the interior of buildings and with no direct natural ventilation to the out-of-doors and which have less than 5 per cent and more than 2½ per cent of the gross floor area of the room, represented by net window and ventilating openings opening to the building in which they are located, and if the net area of all windows and ventilating openings to the out-of-doors of that floor of the building plus the office, rest and lunch rooms is not less than 5 per cent of the gross floor area, no mechanical ventilation shall be required. However, if the net open area of all windows and ventilating openings to the out-of-doors on the floor of the building containing office, rest and lunch rooms is less than 5 per cent of the gross floor area of the floor of the building including office, rest and lunch rooms, then there shall be mechanically supplied not less than 1 cfm of clean tempered air per sq ft of gross floor area to such office, rest and lunch rooms, and there shall be mechanically exhausted 1 cfm of air per sq ft of gross floor area.

"Where a room is used for kitchen purposes only, for the preparation of food for employees, there shall be mechanically exhausted 4 cfm of air per sq ft of gross floor space, and if the net open area of windows and ventilating openings opening to the outside is less than 3 per cent of the gross floor area of such kitchen, there shall also be mechanically supplied 1.2 cfm of clean tempered air per sq ft of gross floor area.

"If the amount of net open area of windows and ventilating openings opening to the out-of-doors is less than 5 per cent of the gross floor area of toilet and locker rooms, there shall be mechanically exhausted 1.5 cfm of air per sq ft of gross floor area."

**Scene in the aluminum foundry at the Dodge Chicago plant, division of Chrysler Corp., which is in production on 18 cylinder, 2200 hp engines for the B-29 and other aircraft**

# WPB...OPA PAW...Etc.

## COAL AND STOKERS

ANTHRACITE DISTRIBUTION CONTROLLED: Solid Fuels Administrator Ickes announced March 31 issuance of regulation to assure that each retail coal dealer will receive fair share of anthracite which will be available for domestic distribution in 1944-45 "coal year" in order that dealers, in turn, will have hard coal to distribute fairly to consumers. The regulation implements regulation governing and limiting retailer-consumer distribution issued the previous week. Both regulations became effective at 12:01 a.m. on April 1, first day of 1944-45 "coal year."

APPROVAL OF CLASS A STOKERS: Regional offices of WPB will process and approve *all* form 1319 applications for class A stokers where the consumption is 25 tons or more annually, irrespective of type of building or boiler in which stoker is installed, it is understood from information received by the Stoker Manufacturers Association.

CERTAIN BITUMINOUS DELIVERIES SET: Solid Fuels Administrator Ickes announced March 31 that retail coal dealers in western states buying bituminous and subbituminous coals shipped by rail from mines in state of Utah must accept shipments of one-twelfth of their annual requirements each month throughout 1944-45 "coal year", which began on April 1.

EARLY BITUMINOUS STORAGE URGED: Industrial consumers of bituminous coal, particularly war plants, have been urged by WPB Chairman Nelson to buy and store maximum amounts of coal during coming months in anticipation of serious deficit in supply for the 12 months starting April 1.

MIDWESTERN COAL USERS: Coal consumers in the midwestern states who delay ordering midwestern-produced coals in hope that sufficient southern Appalachian coals can be made available to them later on are risking a very grave danger of having to get along next winter with considerably less coal than they will need, Solid Fuels Administrator Harold L. Ickes advised.

## EQUIPMENT AND SUPPLIES

BOILER MEN APPROVE WPB-2645: At recent meeting of boiler manufacturers industry advisory committee of WPB, members present were unanimous in recommending continued use of form WPB-2645 (application for authority to purchase from manufacturer), saying that it was of great help to industry in handling production schedules in orderly fashion, WPB said.

CONDENSERS CONTROL RELAXED: Acting to improve performance and reduce man hours of labor in fabrication and maintenance of coil or tube assemblies for refrigeration condensers or coolers, WPB removed restrictions on both use of seamless steel tubing and wall thickness of tubing permitted. (Limitation order L-126, as amended.)

GRILLE SUPPLY SHORT: Registers and grilles for warm air heating systems are in short supply, according to the warm air register and grille industry advisory committee, WPB announced.

INDUSTRIAL INSTRUMENT PROBLEM STUDIED: The industrial instrument industry advisory committee notified WPB representatives that essential commitments of this industry directly and indirectly related to specified urgent war programs could not be fulfilled unless consideration is accorded irreplaceable skilled technicians and experienced engineers in industry.

PRESSURE VESSELS PRICED: Sales of used pressure vessels and enclosed atmospheric pressure vessels which are installed underground, and purchased for use in their present location, have been exempt from price control since April 7, OPA announced. (Amendment 3 to MPR 165 and amendment 113 to MPR 136.)

REFRIGERANT GAS "F-22" RESTRICTED: WPB acted to sharply restrict deliveries of refrigerant gas "F-22", particularly for use in any system of comfort air conditioning and any installation for storing or dispensing carbonated or malt beverages. (Order M-28-a.)

REFRIGERATION ADVISORY FORMED: Office of industry advisory committees, WPB, announced formation of general refrigeration and air conditioning industry advisory committee. Sterling F. Smith is the government presiding officer.

STEEL ENCLOSURES PROVIDED: To provide greater protection for electric control equipment and eliminate ordering of explosion and dust tight enclosures when not absolutely necessary, WPB lifted restrictions on use of steel for general purpose and semi-dust tight enclosures for floor mounted control equipment. (Limitation order L-250, as amended.)

SCHEDULING TABLES REVISED: Table 14 to general scheduling order M-293, listing items under scheduling jurisdiction of WPB's plumbing and heating division, has been amended to bring it into conformity with provisions of that order as amended February 10. The February amendment was designed to simplify the operation of the scheduling procedure and eliminate a substantial amount of paper work. The table, as amended, includes a new column which indicates the number of calendar months for which a delivery schedule of class X products may be frozen under the terms of the order. This action is taken by table 14, as amended April 8, to general scheduling order M-293. The class X products listed in table 14 of M-293, all frozen for two months, are: Metal pipe fabricated beyond rolling mill shapes for resale to installers, excluding pipe only threaded, but-to-length or bevelled by the manufacturer; boilers and boiler units, excluding steel low pressure types to carry less than 15 psi; steel boilers for pressures over 15 psi of water tube, but with under 500 sq ft of heating surface, of water tube, Scotch marine, horizontal return tubular, refractory lined firebox and oil country types; and steel boilers of all sizes to carry over 15 psi of pressure of the steel firebox, vertical and miniature types. Table 5, covering items under scheduling jurisdiction of building materials division, and table 11, covering items scheduled by shipbuilding division, which accompany general scheduling order M-293, also have been revised to conform to the February 10 revision of M-293, which became effective March 1. A new column is added to each table indicating the number of months for which delivery schedule of manufacturers of class X products covered by the tables may be frozen.

STERILIZERS: War Production Board. Restrictions on use of copper and copper base alloys in sterilizer equipment were eased by WPB. (Order L-266, as amended.) These alloys may now be used without restriction in all types except non-pressure instrument sterilizers.

VALVE DISTRIBUTION CONTROL MODIFIED: To effect more balanced distribution of valves and pipe fittings to various war programs, WPB issued direction 1 to table 11 (shipbuilding division table) of general scheduling order M-293.

VALVE MODIFICATION PROPOSED: Proposed changes in specifications for valves and valve parts contained in limitation order L-252 have been recommended by W. B. Holton, Jr., production consultant for valves and fittings of WPB shipbuilding division, following meetings with

members of valve industry. The proposed changes would save production man hours and result in better products with consequent reduction in parts replacements, industry members believe.

WATER HEATERS EASED FOR DAIRIES: Dairy owners will find it easier to purchase water heaters as result of WPB action permitting unrestricted sale of various water heaters used to great extent in dairies, WPB announced. (L-79, as amended.)

WRENCH SPECIFICATIONS RELAXED: Because specified types and sizes of wrenches are required constantly by the armed services and their contractors, restrictions on production and distribution of these wrenches have been relaxed, WPB announced. (Appendix A, attached to schedule II, as amended, to limitation order L-216.)

### FUEL OIL

FUEL OIL "HARDSHIP RATIONS" PROVIDED: Non-residential fuel oil users such as churches, war factories, office buildings, and schools that have done all they could to save oil but have, nevertheless, run out of oil for heating or hot water are now able to obtain "hardship rations". (Amendment 2 to revised RO-11, effective April 15.)

FUEL OIL APPLICATIONS IN JUNE: Simple re-application forms for next season's fuel oil rations will be mailed out to all consumers beginning early in June, Administrator Bowles of OPA said, in announcing details of next season's fuel rationing program.

FUEL OIL YEAR ENDS AUGUST 31: Present fuel year will end August 31 and new year will begin September 1, 30 days earlier than originally scheduled, OPA announced, with result that currently valid "period" fuel oil coupons will be good for deliveries only until August 31, instead of September 30. Next winter's coupons will remain valid for the entire year.

WEST COAST FUEL OIL SHORT: Acting Petroleum Administrator Davies announced that oil-to-coal conversion campaign among large industrial fuel oil consumers in Pacific Northwest must be undertaken immediately to meet rising military demands for fuel oil from west coast ports.

### MODERNIZATION AND REPAIR

FHA FUEL CONSERVATION PROGRAM: Amendment to title 1 of the national housing act permits the making of class 1(a) loans to finance the cost of fuel conservation work performed during the coming months, with the provision that the initial payment be as late as November 1, 1944, Ernest P. Jones Jr., assistant commissioner, announced.

HEATING INSTALLATION RESTRICTED: WPB interpreted order L-79, plumbing and heating equipment, as prohibiting substitution of one type of heating system for another if substitution requires replacement of usable distribution system. This interpretation was issued in answer to questions which have arisen as to whether L-79 permitted the substitution of a wornout or damaged furnace by a hot water or steam boiler, since such a change would entail the replacement of the entire distribution system.

REPAIR REGULATION EXPLAINED: Repair services for many domestic appliances are handicapped, WPB said, because many electrical and mechanical repair shops are not taking advantage of WPB regulation designed to aid them in obtaining parts and materials. These items include automatic heating plants and plumbing. (CMP regulation 9A.)

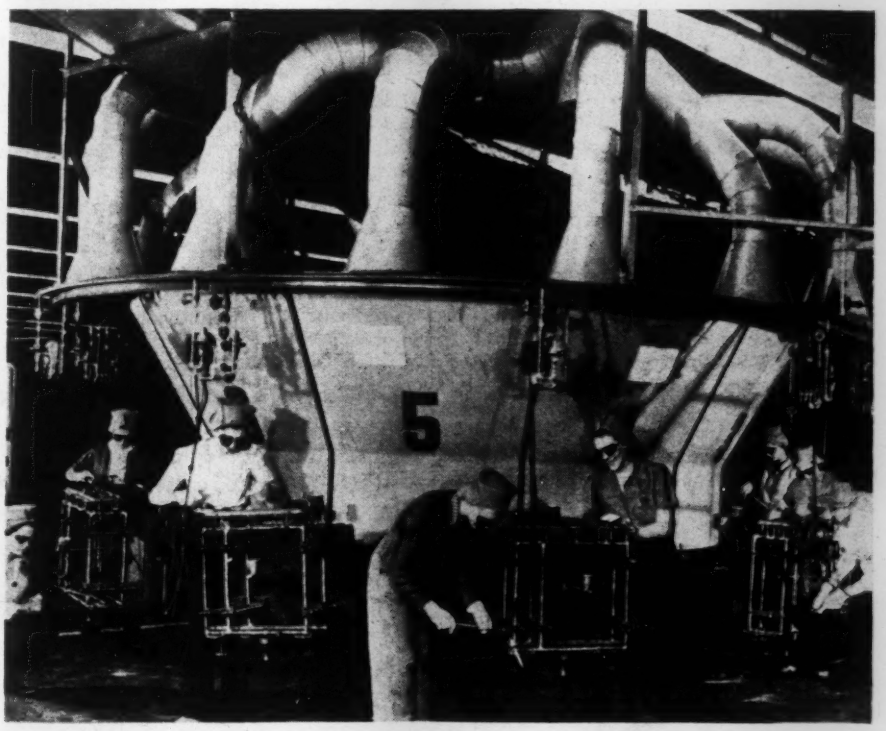
REPAIR SUPPLIES CONTROL MODIFIED: Procedure under which repairmen purchased controlled materials, other materials, parts, and subassemblies, has been amended to indicate that they may use it to obtain materials with which to recondition or rebuild damaged or used items for resale, WPB announced. (CMP regulation 9A as amended.)

### WAR HOUSING

WAR HOUSING PROCEDURE MODIFIED: Contractors on publicly financed war housing projects will furnish in future items of equipment heretofore handled through mass purchase by Federal Public Housing Authority, Commissioner Herbert Emmerich announced. Such items as plumbing and heating equipment and lighting fixtures will be included in construction contracts again as they are under normal building conditions.

## PRODUCTION MERRY-GO-ROUND

**The designer of this rotating assembly line containing frames for radio sets must have been inspired by the oldtime merry-go-round. The frames rotate around 10 work stations, while the center section and ducts above carry off fumes due to the brazing of the metal parts (Photo, Acme Newspictures, Inc.)**



## • MEETINGS & CONVENTIONS •

*Apparently June is not only the month most popular with brides, but also with those who make arrangements for meetings and conventions. Here's the schedule of such affairs for coming weeks of interest to HPAC's readers:*

### Fuel Oils Joint Conference

Fuel Oils Commercial Standard CS12-40, Assumptions and Facts Joint Conference, 10 a.m., Thursday, May 18, Palmer House, Chicago, Ill. Refiners and oil burner manufacturers are to meet to consider a draft of assumptions and basic facts concerning the production and use of fuel oils so that plans may be made for changes in oils and burners necessitated by current and projected changes in refinery practices.

### Heating, Piping, and Air Conditioning Contractors

Heating, Piping and Air Conditioning Contractors National Association, 55th annual convention, May 31-June 2, Hotel Statler, Cleveland, Ohio. A condensed, five session program will include committee reports and discussions on training and testing welding operators, formation and operation of welding bureaus, home heating, selling heating, employer-employee relationships, air conditioning, and postwar plans for the industry. There will be an evening round table session Wednesday with Leon L. Munier presiding and with an illustrated talk on panel heating by Charles A. Hawk Jr., A. M. Byers Co., and an afternoon address Thursday by James F. Lincoln, Lincoln Electric Co., on the future of welding. Secretary of the HPACCNA is Joseph C. Fitts, 1250 Sixth Ave., New York 20, N. Y.

### Stoker Manufacturers

Stoker Manufacturers Association, annual meeting, June 1-2, French Lick Springs, Ind. Headquarters office of the SMA is at 307 N. Michigan Ave., Chicago 1, Ill.

### Refrigerating Engineers

American Society of Refrigerating Engineers, 31st annual meeting, June 5-7, William Penn Hotel, Pittsburgh, Pa. Papers are to be given on refrigerants and absorbents, by William R. Hainsworth, Servel, Inc.; hydrocarbon refrigerants in low temperature fields, by H. D. Edwards, Linde Air Products Co.; blast freezer plants, by Earl D. Pollock, Vilter Mfg. Co.; heat flow rates with rapid temperature changes in enclosed space, by H. B. Pownall and S. P. Soling, York Corp.; control of a temperature altitude chamber, by F. W. McKenna, Vilter Mfg. Co.; chromate corrosion inhibitors in brine systems, by Marc Darrin, Mutual Chemical Co. of America; cold cathode fluorescent lighting in refrigerated areas, by R. C. Hultgren, General Luminiscent Corp.; the postwar domestic refrigerator, by Arthur W. Ewell, Worcester Polytechnic Institute; industrial plastic materials, by John Sasso, Product Engineering; refrigeration in the synthetic ammonia industry, by J. G. Doly, consulting engineer; psychrometry in the frost zone, by D. D. Wile, Carrier Corp.; determining thermal conductivities at low temperatures, by Ludwig Adams, Mellon Institute; and refrigeration test equipment, by Robert W. Christie, U. S. Testing Co. Headquarters office of the ASRE is at 50 W. 40th St., New York 18, N. Y.

### Smoke Prevention

Smoke Prevention Association of America, annual meeting, June 6-9, Hotel Statler, Detroit, Mich. Program includes addresses on the activities of the national fuel efficiency section of the U. S. Bureau of Mines, by Thomas Cheasley; the postwar influence on smoke abatement of the bituminous coal research program, by Julian Tobey; locomotive smoke abatement in terminals, by P. [illegible] Tolin; and a round table discussion on railroad smoke a[illegible]ement and locomotive fuel problems, with Eugene D. B[illegible]ton as moderator. Headquarters office of the SPA is at 139 N. Clark St., Chicago 2, Ill.

### Warm Air Heating

National Warm Air Heating and Air Conditioni[illegible] Association, mid-year meeting, June 7-8, Stevens Hotel, [illegible]hicago, Ill. A. P. Kratz, University of Illinois, will discu[illegible]s principles of smokeless furnace operation, S. Konzo, U[illegible]versity of Illinois, will present a new forced warm air ma[illegible]al and outline a warm air textbook, Ralph Sherman, Battel[illegible] Memorial institute, will discuss fuel resources, and talks on gas heating and oil heating will be given. Numerous other subjects will be considered. George Boeddener is managing director of the NWAHACA, and the headquarters office is at 145 Public Square, Cleveland 14, Ohio.

### District Heating Association

National District Heating Association, 35th annual meeting, June 14-15, William Penn Hotel, Pittsburgh, Pa. Program includes reports of officers and committees and the papers to be given include discussions on sizing economizers and condensate coolers, by W. A. Schulmeister, American District Steam Co.; steam consumption for water heating, by A. F. Metzger, Allegheny County Steam Heating Co.; sprinkler tank water temperature control, by George H. Tuttle, The Detroit Edison Co.; correlation of steam use in buildings and use of district steam in certain types of air conditioning, by A. R. Mumford; degree day standardization, by John F. Collins Jr.; studies on use of amines in control of corrosion in steam heating systems, by A. A. Berk, U. S. Bureau of Mines; results of corrosion research work at Carnegie Institute of Technology, by D. S. McKinney, of Carnegie Institute; corrosion in steel coal bunkers; problems in use of inferior wartime coals; use of concrete poured directly around an underground steam line; cell concrete jobs, by Mr. Schulmeister; installation of a 1000 ft temporary high pressure main to an industrial user; lengthening the life of underground steam lines, by A. A. Sellke, The Detroit Edison Co.; electrolysis of underground steam mains; experience with a diaphragm operated relief valve, by G. T. Siebenthaler, Dayton Power and Light Co.; inspection, testing, and reading of steam meters, by Henry L. Warhanek; changes in testing and maintenance of condensation meters due to wartime conditions, by Richard H. Brown, Rochester Gas and Electric Corp.; effect of testing meters biennially or triennially instead of annually on meter accuracy, by Mr. Warhanek; and others. R. J. J. Tennant, National Association of Building Owners and Managers, will give a talk entitled From the Other Side of the Fence. John F. Collins Jr., 827 N. Euclid Ave., Pittsburgh, is secretary-treasurer of the NDHA.

### Heating and Ventilating Engineers

American Society of Heating and Ventilating Engineers, semi-annual meeting, June 19-20, Pantlind Hotel, Grand Rapids, Mich. See ASHVE Journal Section in this issue for further information. Secretary of the ASHVE is A. V. Hutchinson, 51 Madison Ave., New York 10, N. Y.

### Mechanical Engineers

American Society of Mechanical Engineers, semi-annual meeting, June 19-22, William Penn Hotel, Pittsburgh, Pa. Problems of industry, production, transportation, and management will be discussed and many inspection trips to modern plants will be arranged. Headquarters office of the ASME is at 29 W. 39th St., New York, N. Y.

### Power Engineers

National Association of Power Engineers, national convention, August 22-August 25, Hotel Jefferson, St. Louis, Mo. F. C. Laufketter, Hotel Jefferson, St. Louis, is general chairman of the local convention committee.

Heating, Piping & Air Conditioning

AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

Journal Section

H. D. Bratt

T. D. Stafford

W. W. Bradfield

C. H. Pesterfield

# Semi-Annual Meeting 1944

## June 19-20 — Grand Rapids

PREPARATIONS FOR the Semi-Annual Meeting 1944 are well advanced and Western Michigan Chapter members look forward to the welcoming of a large number of visiting members who will attend this wartime meeting of the Society at the Pantlind Hotel, Grand Rapids, Mich., June 19-20.

In advance of the regular technical sessions there will be a meeting of the Council, the Committee on Research, and of many of the Research Technical Advisory Committees.

Papers offered for this meeting include the Engineering Control of Solvent Hazards, Air Sterilization and Disinfection, Heat Pump Tests, Weather Data, Effect of Attic Ventilation on Comfort and Fuel Conservation Studies.

The Program Committee plans for three technical sessions and has selected papers that are important to the war production industries.

Chairman T. D. Stafford, of the Committee on Arrangements, states that every precaution is being taken to provide for the comfort of members, but because of present conditions faced by hotels and transportation lines it is important to:

1. Make hotel reservations 30 days in advance and not later than June 10, so that the headquarters hotel can provide the kind of rooms that are requested.
2. It is important to indicate the approximate hour of arrival, and if original date is changed advise the hotel promptly.
3. When making reservations members should use the special card provided to indicate their identity with the ASHVE.
4. Railroad accommodations and tickets should be purchased 30 days in advance of the meeting, and Pullman accommodations for both going and return trips should be made before leaving home.
5. Whenever possible travel on off-hour trains and allow adequate time for connections where through car service is not available.
6. Travel with the minimum amount of personal baggage.

Grand Rapids is served by Pennsylvania Central Air Lines, New York Central System, the Pennsylvania Railroad, Pere Marquette and Grand Trunk Railway.

The Committee on Arrangements have been announced as follows: W. W. Bradfield, *Honorary Chairman;* T. D. Stafford, *General Chairman;* C. H. Pesterfield and H. D. Bratt, *Vice-Chairmen.* The chairmen of the committees are as follows: *Dinner*—F. C. Warren; *Inspection Trips*—Arthur Boot; *Ladies*—Mrs. O. D. Marshall; *Finance*—Karl Ziesse; *Special Events*—O. D. Marshall; *Golf*—W. G. Schlichting; *Reception*—L. G. Miller; and *Publicity*—Vaughn Hill.

1894 A GREAT PAST A GREATER FUTURE 1944

# Some Effects of Attic Fan Operation on Comfort

By W. A. Hinton,* Durham, N. C. and W. G. Wanamaker,** Atlanta, Ga.

**This paper is the result of research sponsored jointly by the Atlanta Chapter and the Committee on Research of the American Society of Heating and Ventilating Engineers in cooperation with the Engineering Experiment Station at the Georgia School of Technology.**

**SUMMARY — When cool night air was drawn through a single story frame structure at a rate of 45 air changes per hour, inside air temperature was reduced to within two degrees of outside air temperature. Air velocities ranging from 18 feet per minute to 169 feet per minute were observed. Mean radiant temperature was found to average approximately one degree above inside air temperature. Effective temperatures in the comfort zone were obtained early in the evening, from 7:00 to 8:30, depending on outside air temperatures and air velocities used in the house.**

TO MEASURE the effect of attic fan operation on comfort, an investigation was started by the Atlanta Chapter and the Committee on Research of the Society in cooperation with the State Engineering Experiment Station at the Georgia School of Technology. In a former report the results obtained in the summer of 1941 were presented to the society.[1]

The data gathered came from tests conducted in two practically identical houses, one of which was equipped with an attic fan. Findings were reported on the effect of various air changes per hour on air temperature. Wet-bulb temperatures were noted and a few measurements of air velocity were made, but these data were so meager that they were not included in the report. It was decided to extend the investigation during the summer of 1942; the additional results are reported here.

## Test Equipment and Instruments

### *The House*

The two houses used during the summer of 1941 were not available for further use; consequently another Atlanta house was selected. It was a single story frame structure without a basement. The windows were double-hung wood sash, without awnings. The interior walls and ceilings were of wood, lath, and plaster. Fig. 1 is a floor plan of this house, showing the location of test instruments.

### *The Fan*

The fan used was a four bladed axial or propeller type, 35 in. in diameter. It was rated to deliver 10,000 cfm at a speed of 350 rpm when operating against a static pressure of 0.01 in. of water. This fan was installed in the house in such a manner as to draw air through an opening in the ceiling of Room IV. The fan discharged air into the attic space. The gross area of the opening in the ceiling was 17.5 sq ft and was covered by an expanded metal grille having 70 per cent free area. The net areas of openings by which air left the attic totaled 25 sq ft.

### *Pyrometer*

The air temperatures were measured with iron-constantan thermocouples. Because of the limited personnel, the thermocouples were attached to a sensitive potentiometer pyrometer of the automatic recording type, in order to study the complete daily temperature cycles. All air temperature measurements were made 30 in. above the floor level. The points at which temperatures were observed are shown in Fig. 1 by means of cross (+) marks. In addition, temperatures of outside air and of air in the attic space were recorded.

Wet- and dry-bulb temperatures of the inside air for determination of relative humidity were recorded on a separate instrument.

### *Anemometers*

Velocities of air at points such as open windows were measured with a calibrated 4-in. anemometer. For studying air velocities at other points throughout the house *heated thermocouple anemometers*[2] of the type developed by Kratz, Hershey, and Engdahl were used. Recently a calibration table for this anemometer, covering a range of 16 fpm to 3000 fpm was supplied by Professor Kratz[3]. Four of these anemometers were connected to a control panel with sufficient lead wire to reach the stations where velocity determinations were desired. The location of stations where air velocity was measured is shown in Fig. 1; each station is indicated by a circle. The velocities were determined at the same level as the temperatures—30 in. above the floor.

### *Globe Thermometer*

A globe thermometer was used to estimate the mean radiant temperature. The globe was constructed from a 6-in. copper sphere. This instrument was similar to the Vernon globe thermometer[4], but the globe temperature was measured by a thermocouple as suggested by Houghten, Gunst, and Suciu.[4] The globe was supported 30 in. above the floor and was located in the center of the room. Further information on the use of this type of instrument has been reported by Bedford and Warner.[5]

## Test Methods

The fan was turned on during afternoons at times ranging from 3 to 5 p.m. and was turned off at approximately 9 a.m. the following morning. (Had a time clock been available the fan would have been

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[1]Exponent numerals refer to Bibliography.

**Note:** Published by permission of the Director, State Engineering Experiment Station.

For presentation at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Grand Rapids, Mich., June 1944.

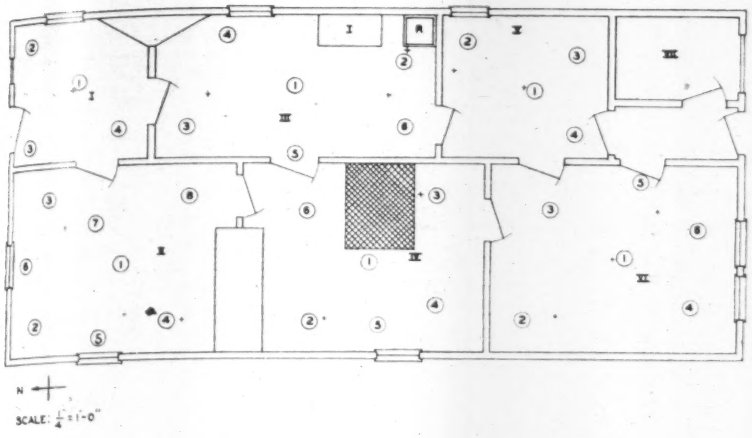

**Fig. 1—Floor plan, showing location of instruments**
**+ Air temperature measuring station**
**O Air velocity measuring station**
**I Instrument table**
**R Recording Pyrometer**

turned off somewhat earlier in the morning.) During periods when the fan was operating the windows were kept open and the air temperatures were obtained from the charts of the recording pyrometer.

The procedure used for obtaining mean radiant temperatures was to place the globe in the center of the room 30 in. above the floor and to record temperature inside the globe, air temperature outside of the globe not more than one foot from the globe. Air velocity at the same level as the globe, and not more than one foot from it, was recorded. Most of these measurements were made with the fan in operation, since it was desired to determine the effect of fan operation on mean radiant temperature.

The air movement and distribution was studied by using two or three procedures, each of which was somewhat similar to the others. One procedure was to put the fan in operation and determine average velocities at open doors and windows with the four-inch anemometer. The averages were obtained by the usual method of dividing the area into stations, taking one-minute readings of anemometer at each station, then adding the velocities, and dividing by the number of stations to obtain the average value.

The second procedure was to have the fan in operation and determine air velocity by means of a heated thermocouple anemometer at each station indicated on Fig. 1. During these tests the windows and doors of all rooms were opened.

The third procedure was the same as the second except that certain rooms were closed and others left open. The air velocity was then measured at all the stations in the rooms that were left open. In this procedure there were several possible combinations of rooms that could be left open. Obviously, Room IV, in which the fan grille was located, had to remain open in all cases. The following combinations were studied:

Rooms II and IV open; Rooms IV and VI open; Rooms III and IV open; Rooms II, IV, and VI open; Rooms III, IV, and V open; Rooms IV, V, and VI open; and Rooms I, III, and IV open. Air velocities were measured with the heated thermocouple anemometers.

## Results and Discussion

Air temperatures were recorded on 15 days. The curves shown in Fig. 2 are typical of the results obtained. These curves show the variation of both inside and outside air temperature over a 24-hour period. During this test the capacity of the fan was 45 air changes per hour based on the total volume of the house; all rooms were open. These curves confirm the results previously obtained in the earlier investigation.[1] It will be noted that while the fan was operating the difference between the inside and outside air temperature did not exceed three degrees Fahrenheit. Most of the time the temperature differential was not over two degrees Fahrenheit.

The results of the tests in which air movement and distribution were studied are shown in Tables 1, 2, and 3. Although the fan intake grille was located near the center of the house as shown in Fig. 1, there were large variations in air velocities and in quantities of air entering the several windows of the house. Table 1 shows the average velocity at each door and window when the fan was in operation. It may be seen that in Room II the air velocity was 63 fpm at the north window, but was only 20 fpm at the west window. It is also interesting to note that the highest velocity was not obtained at the window of Room IV, the room in which the fan grille was located, but at the window of Room III.

The distribution of air in terms of cubic feet per minute and air changes per hour is given in Table 2. These quantities are based on air actually entering the rooms from the outside and do not include any air that may have passed through one room and then into another. The largest volume of air entered through the window of Room III, while the next largest entered through Room I. From this table it can be seen that the air changes per hour varied from 20 for Room II to 107 for Room I.

**Table 1—Average Velocities at Windows and Doors**

| Room No. | I | IIN | IIW | III | IV | V | VI |
|---|---|---|---|---|---|---|---|
| Velocity at Windows, Ft/Min..... | 144 | 63 | 20 | 291 | 188 | 48 | 67 |
| Front Outside Door—30 Ft/Min. | | | | Back Outside Door—22 Ft/Min. | | | |

Fan on—all windows and doors open N-North W-West

**Table 2—Distribution of Air Flow**

| Room No. | Volume of Room Cu Ft | Air Flow CFM | Air Flow Per Cent Total | Air Changes Per Hour |
|---|---|---|---|---|
| I | 894 | 1595 | 22.8 | 107 |
| II | 1904 | 635 | 9.1 | 20 |
| III | 1664 | 1965 | 28.2 | 71 |
| IV | 1904 | 1440 | 20.6 | 45 |
| V | 954 | 605 | 8.7 | 38 |
| VI | 2064 | 745 | 10.6 | 27 |
| Total | 9384 | 6985 | 100.0 | 45 Avg. |

Above values based on air entering from outside.

**Table 3—Air Velocities Feet per Minute**

| Room No. | I | | | | II | | | | | | | | III | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Station No. | 1 | 2 | 3 | 4 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 1 | 2 | 3 | 4 | | |
| Rooms Open: All | 169 | 42 | 112 | 56 | 45 | 23 | 18 | 27 | 73 | 88 | 52 | 45 | 156 | 37 | 59 | 89 | | |
| Rooms Open: II-IV | .. | .. | .. | .. | 158 | 16 | 18 | 47 | 166 | 145 | 82 | 82 | .. | .. | .. | .. | | |
| Rooms Open: IV-VI | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | | |
| Rooms Open: III-IV | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | 30 | .. | 44 | 240 | 62 | 87 |
| Rooms Open: II-IV-VI | .. | .. | .. | .. | 144 | 16 | 27 | 43 | 122 | 97 | 61 | 61 | .. | .. | .. | .. | | |
| Rooms Open: III-IV-V | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | 122 | 52 | 159 | 317 | 43 | 57 |
| Rooms Open: IV-V-VI | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | | |
| Rooms Open: I-III-IV | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | 51 | 36 | 266 | 79 | 65 | 35 |

| Room No. | IV | | | | | | V | | | | VI | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Station No. | 1 | 2 | 3 | 4 | 5 | 6 | 1 | 2 | 3 | 4 | 1 | 2 | 3 | 4 | | |
| Rooms Open: All | 104 | 81 | 104 | 68 | 104 | 104 | 44 | 76 | 19 | 45 | 45 | 43 | 43 | 28 | | |
| Rooms Open: II-IV | 228 | 28 | 156 | 188 | 107 | .. | .. | .. | .. | .. | .. | .. | .. | .. | | |
| Rooms Open: IV-VI | 354 | 57 | 210 | 28 | 210 | 28 | .. | .. | .. | .. | 157 | 84 | 41 | 157 | 66 | 39 |
| Rooms Open: III-IV | 88 | 34 | 35 | 35 | 226 | 51 | .. | .. | .. | .. | .. | .. | .. | .. | | |
| Rooms Open: II-IV-VI | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | 124 | 35 | 35 | 70 | 125 | 137 |
| Rooms Open: III-IV-V | 57 | 75 | 41 | 46 | 105 | 181 | 118 | 81 | 39 | 65 | .. | .. | .. | .. | | |
| Rooms Open: IV-V-VI | 167 | 85 | 300 | 85 | 230 | 164 | 81 | 115 | 43 | 141 | 32 | 40 | 81 | 32 | 46 | 32 |
| Rooms Open: I-III-IV | 83 | 40 | 48 | 48 | 88 | 111 | .. | .. | .. | .. | .. | .. | .. | .. | | |

Velocities measured with heated thermocouple anemometers—30" above floor.

The differences in air volumes and air changes per hour were due, in part, to the resistance to air flow offered by various rooms. In general those rooms farther from the fan inlet would offer greater resistance and consequently have less air flow through them. Another factor that influenced the air entering various rooms was that a hill directly west and to the south of the test house sheltered the windows of Rooms II and IV from wind coming from that direction. The windows of Rooms I, III, and V were more favorably exposed to wind since the space east of the house was vacant; the nearest building was 200 ft away. A house located 40 ft south of the test house affected somewhat the amount of air entering the window of Room VI. In comparing air changes per hour the volumes of the rooms should be kept in mind. Since small rooms often have windows as large as the other rooms, the volume of air entering a small room may equal that entering a larger room; but, in such a case, the smaller room has a larger number of air changes per hour due to its smaller volume.

A careful check of the temperature data was made to ascertain if the large difference in air changes made an appreciable difference in temperature. It was found that the difference in temperature between rooms never exceeded one degree Fahrenheit.

The air velocities existing at various points in the house are shown in Table 3. With the fan in operation and all windows open, velocities from 18 fpm to 169 fpm were produced. A check of Table 3 against Fig. 1 will reveal the fact that most of the lower velocities were at points such as corners, where a rather small degree of air movement would be anticipated. It is important to note that while some locations had velocities much below the average, there was sufficient turbulence in the air flow to produce a reasonable degree of air movement even in corner locations. Table 3 shows also that the higher velocities occurred in rooms having greater number of air changes per hour.

Table 3 shows in addition the increase in air velocity that occurs in certain rooms when the doors and windows of other rooms are closed. It may further be noted that certain combinations of rooms were more effective in increasing air velocity than others.

A few apparently erroneous velocities in Table 3 may be explained by pointing out that the closing of doors in the house places some of the velocity stations in corners, and it also forces the air movement to occur along a somewhat different path.

The mean radiant temperatures were computed from the observed data by means of the Bedford-Warner formula.[5] An examination of a large number of readings taken on five different days showed that mean radiant temperature was always very near room air temperature; the difference never exceeded three degrees. During certain periods of time, mean radiant temperature was equal to room air temperature. In the evenings when cool night air was being drawn through the house, mean radiant tempera-

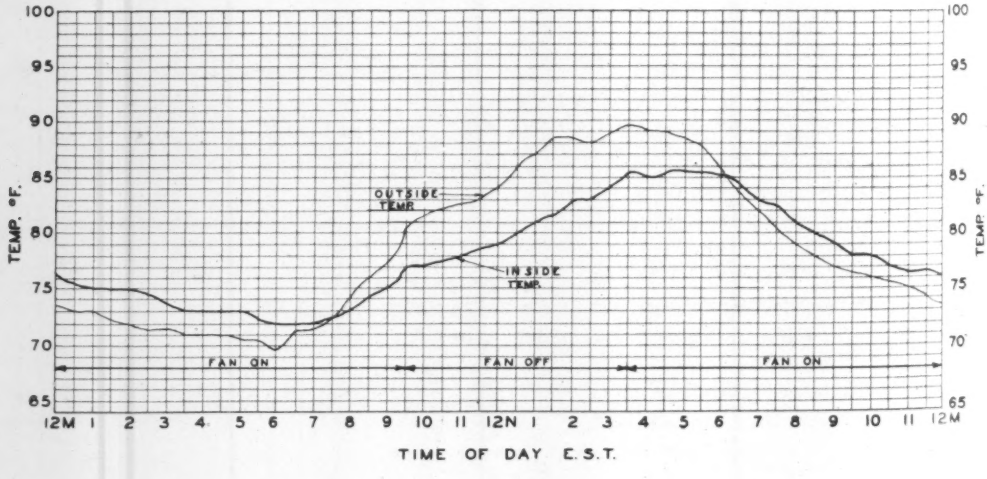


**Fig. 2—Relation between the time and inside and outside temperatures. 45 air changes per hour 9-18-42**

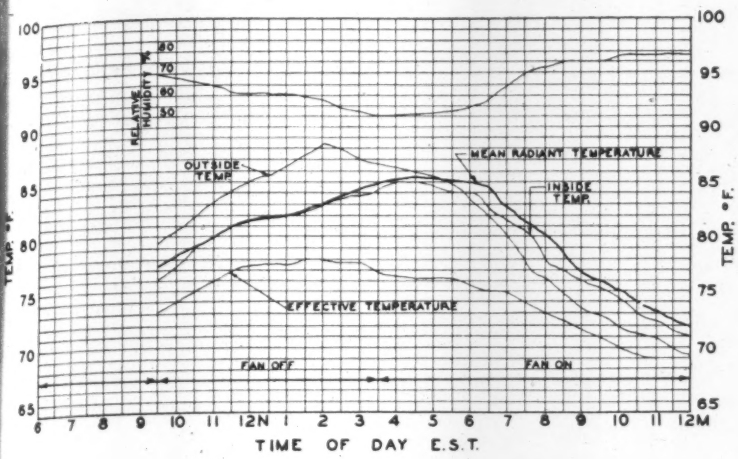


Fig. 3—Curves of time and comfort factors with 45 air changes per hour 9-11-42

ture was only slightly higher than air temperature. A great part of the time the difference was not over one degree. The results on mean radiant temperature are presented in the same manner as other temperatures. The curve of Fig. 3 is typical of the values observed.

This curve shows that attic fan operation reduces mean radiant temperature at approximately the same rate as it reduces room air temperature. The curve in Fig. 3 is for Room VI. This room was chosen because it was the southwest corner room and was expected to be affected by solar radiation during the afternoon to a greater extent than some of the other rooms because it was exposed to the sun in the afternoon.

An effective temperature curve for Room VI is shown in Fig. 3. This curve follows the same trend as the inside air temperature curve. It is quite evident that when the fan was turned on the increase in air velocity produced an immediate drop of approximately one degree effective temperature.

An effective temperature of 73 deg is recommended [6, 7] for summer comfort in the southeast portion of the United States when the dry-bulb temperature ranges from 91 deg to 95 deg. The effective temperature curve of Fig. 3 shows that this value of 73 deg was reached at 8:00 p.m. (E.S.T.) at which time the dry-bulb temperature was 78 deg and that it is reduced still further as the outside temperature decreases. (The values of effective temperature were obtained from a comfort chart.[8])

In addition to the effective temperature curve of Fig. 3, the effective temperatures that would have existed in Room VI with various velocities have been plotted in Fig. 4.

## Discussion and Conclusions

The mean radiant temperatures reported here are in agreement with what might be expected. During periods when the fan was off, transfer of heat between surfaces and air would tend to equalize the temperature of air and surfaces. During periods when the fan was on, the mean radiant temperature was only slightly above air temperature. Previous studies of night air cooling [1, 10] have shown that in a frame house, surface temperatures are not more than a degree or two above room air. The surface temperatures reported have for the most part been ceiling surfaces, so the average would be less in some cases. Therefore, it would be impossible for mean radiant temperature to exceed air temperature by more than one or two degrees in a frame house with this type of cooling.

It is possible that the mean radiant temperatures reported may not be the exact values for the time given, because of the time lag of the globe thermometer. The measurement of mean radiant temperature by the globe thermometer is based on the globe's coming into thermal equilibrium with its environment. In these tests the air temperatures and wall surface temperatures were continuously changing. Therefore, it is unlikely that true thermal equilibria were reached or that the mean radiant temperatures reported are exact. However, it seems reasonable to believe that the values given represent the magnitude and variation in mean radiant temperatures with good accuracy even though they may be slightly displaced as to time.

The results presented give some basis for a discussion of the proper number of air changes per hour. The attic fan used in a given installation should have sufficient capacity to produce the desired comfort conditions as early in the evening as possible.

From the viewpoint of air temperature, not more than 30 or 40 air changes per hour would be economical. Nevertheless, air velocity

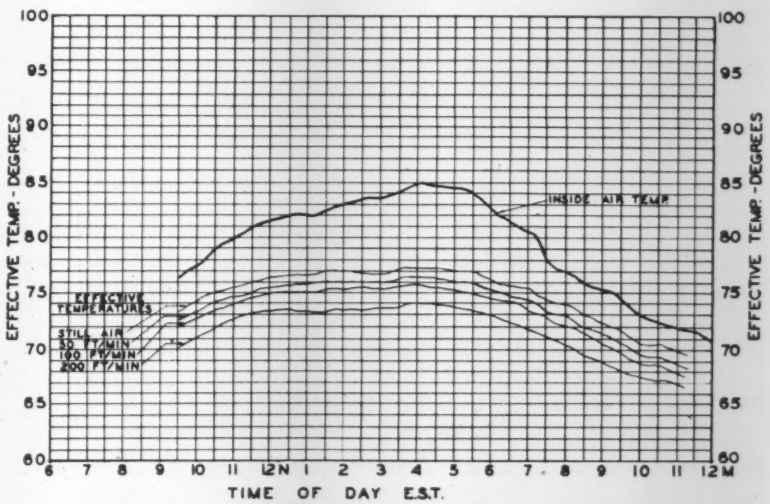


Fig. 4—Curves of time and effective temperatures at various velocities 9-11-42

is a factor in comfort, and the selection of the number of air changes to use must take velocities into account. Fig. 4 shows the effective temperatures that would have existed in Room VI had the air attained the velocities indicated on the curves. When all rooms were open, Room VI had average of velocities of 41 fpm; thus it can be seen that comfort conditions of 73 deg ET were reached by 8 p.m. This room had 22 air changes per hour. For the same conditions Room III had 70 air changes per hour, and the velocities averaged 74 fpm, therefore, from Fig. 4 it is seen that Room III would have reached 73 deg ET at 7:30 p.m. or 30 min earlier. Using the same two rooms but selecting maximum rather than average velocities the results are: 73 deg ET reached by Room VI at 7:45 p.m. and by Room III at 6:45 p.m., or the comfort temperature is reached one hour earlier.

While these data indicate the desirability of high air velocities it is of interest to note a simple method of obtaining higher air velocities without the expense of an extra large fan. Table 3 reveals that when only Rooms IV and VI were open, Room VI had an average velocity of air of 124 fpm and would have reached 73 deg ET earlier than either of the previous illustrations.

The foregoing facts and illustrations indicate that while 30 to 40 air changes per hour are sufficient for reducing the inside air temperature to within two degrees of the outside temperature, it is desirable to produce higher air velocities so as to lower the effective temperature. The authors are of the opinion that satisfactory results may be obtained if fans are installed which are capable of producing 40 air changes per hour instead of the generally accepted standard of 60 air changes per hour. (These air changes are based on the total volume of the livable space in the house, and refer to actual air delivered and not to the maximum rated capacity of the fan.) At times when the outside air temperature is unusually high, the greater air velocities desired may be obtained by the simple expedient of closing the rooms which are not occupied. Attic fans are rated at free delivery. Flow resistances encountered in actual installations cause the actual air flow to be less than the rated capacity of the fan.

Measurements made at the intake grille show that the fan used in these tests delivered from 70 to 75 per cent of its rated capacity. Larger intake grilles or another type of grille might increase this percentage.

It is further suggested that means be provided to reduce air movement when comfort has been attained as an examination of Figs. 3 and 4 will show that what might be a pleasant breeze at 9 p. m. may become an objectionable draft by 12 p.m. unless the fan is stopped or its capacity reduced. Although a variable speed motor and thermostatic controls might be used to accomplish this, the most economical means is to use a time clock which operates a switch to stop the fan motor at a time selected by the occupants of the house.

It is interesting to estimate the difference in effective temperature between a house with attic fan and one without an attic fan. Tests have shown that air temperature in a frame house does not decrease much before 8 p.m. if no fan is used. Fig. 3 shows that 84 F dry-bulb may be assumed for air temperature in the house at 8 p.m. If the air is still, as it would be without a fan, an effective temperature of 80 deg results; this is seven degrees higher than the 73 deg ET attained at 8 p.m. with attic fan in operation.

The small difference between air temperature and mean radiant temperature indicates that this difference is not a large factor in determining fan capacity. Since mean radiant temperature will always be near air temperature it is important to observe that if no fan is used mean radiant temperature will be higher than with a fan and will contribute to the discomfort experienced.

The following is the principal conclusion that may be drawn from attic fan data available. The comfort produced depends primarily on only two things for a given house, the conditions of the outside air and the velocity that the fan produces.

## Summary of Results

The results of tests on a single-story frame house equipped with an attic fan, and located in Atlanta, may be summarized as follows:

1. The inside air was found to be approximately two degrees Fahrenheit above the temperature of the outside air when average number of air changes per hour for the house was 45.

2. With an average for the house of 45 air changes per hour some rooms had as low as 20 air changes per hour while others had in excess of 100.

3. In spite of different rates of air change for various rooms, the air temperature did not vary more than one degree Fahrenheit from one room to another.

4. Large numbers of air changes are useful only because the increased air velocity decreases effective temperature.

5. The average mean radiant temperature was found to be about one degree above air temperature. The difference never exceeded three degrees.

6. With the fan in operation and all doors and windows open, air velocities varied from 18 to 169 fpm. The lower values were obtained in corners.

7. When some of the rooms were closed the velocity of air at points in other rooms showed a marked increase. Increases of 100 per cent were noted in many cases and 200 per cent in a few locations.

8. With the fan in operation during the evening effective temperature decreased at approximately the same rate as outside dry-bulb air temperature.

## Acknowledgment

The authors wish to acknowledge and to express appreciation for valuable assistance to Prof. A. P. Kratz, University of Illinois, who supplied information on use of the heated thermocouple anemometer; to Dr. F. C. Houghten of the ASHVE Research Laboratory who very kindly gave advice on construction and use of the globe thermometer. The cooperation of Prof. R. S. King, head, Department of Mechanical Engineering, who permitted the use of special instruments from his department, is greatly appreciated. Appreciation is also expressed for the splendid cooperation and assistance of T. T. Tucker, Chairman, W. J. McKinney, and Leo Sudderth, members of the Atlanta Chapter's Research Committee of ASHVE. The writers

also appreciate the help given by the other members of the Atlanta Chapter.

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## ASHVE'S FIRST LOCAL CHAPTER FORMED 38 YEARS AGO

During this 50th Anniversary Year it is of interest to note that the Illinois Chapter of the Society—oldest of the Society's local groups and one of the largest and most active—was formed 38 years ago, about twelve years after the national body came into existence. On July 19, 1906, fourteen members of the ASHVE made application at the Semi-Annual meeting of the Society, which was being held in Chicago "to form a local chapter of the Society, headquarters of which should be located in Chicago."

The 14 members who signed the application were T. J. Waters, August Kehm, J. M. Stannard, John F. Hale, R. A. Widdicombe, N. L. Patterson, S. R. Lewis, James Mackay, E. F. Capron, George Mehring, John K. Allen, S. G. Neiler, W. B. Graves, and John Boylston.

The Illinois group really started something, for nothing could be done at the meeting in the way of granting their application until the By-Laws of the Society were amended to permit local chapters. A motion was therefore carried requesting the Board of Governors of the ASHVE to make a mail ballot on the proposition of amending the By-Laws to permit local chapters. Another motion was carried which provided that if the proposed amendment to the By-Laws was made that permission to form a local chapter in Illinois be granted.

The necessary changes in the constitution and by-laws of the ASHVE were made, and on December 3, 1906, nine Society members, residents of Chicago, met and formed the Illinois Chapter. At this meeting they elected the following officers for the year 1907: T. J. Waters, *president*; James Mackay, *vice-president*; John F. Hale, *secretary*; August Kehm, *treasurer*; and George Mehring, Edmund F. Capron and Samuel R. Lewis, *directors*.

The chapter was incorporated on March 5, 1907, with 17 responding to the call and becoming charter members of the Illinois group: W. L. Bronough, Mr. Capron, James H. Davis, Mr. Graves, Mr. Hale, Mr. Kehm, Mr. Lewis, Mr. Mackay, D. F. Morgan, Mr. Mehring, Mr. Patterson, William A. Pope, Mr. Stannard, Mr. Waters and Mr. Widdicombe, with Messrs. Allen and Boylston as associate members. Mr. Hale presented a report for the Illinois Chapter at the Society's semi-annual meeting in July, 1907, at which time the chapter had twenty-three members. Monthly meetings, then as now, were held from October to May each year. In 1907, the meeting rooms of the Western Society of Engineers, in the Monadnock Block, Chicago, were used. In recent years, including this season, the Chapter meets in the quarters of the Merchants and Manufacturers Club, in the Merchandise Mart.

Mr. Lewis, charter member of the Chapter, one of its original directors, later president of the Chapter as well as president of the ASHVE and Life Member of the Society and the Chapter, spoke on hot water heating and boilers at the first of the Illinois Chapter's regular meetings. Mr. Lewis, retaining his active interest in Chapter affairs throughout the years since 1906, led off the Chapter's 1942-43 season by presenting a talk (at the insistence of the Chapter's meetings and publications committee) on adventures in war preparation and production, in which he reviewed some of his experiences as a consulting engineer in the heating, ventilation and air conditioning of war plants and other wartime buildings.

At the time of this writing the oldest chapter of the ASHVE completes its thirty-eighth season of monthly meetings with a membership of 150, seven of whom are serving in the Army or Navy. The officers for the 1943-44 chapter year were C. E. Price, *president*; A. O. May, *vice-president*; K. C. Porter, *treasurer*; and C. M. Burnam, Jr., *secretary*. The *Board of Governors* consisted of E. M. Mittendorff, A. C. King, J. S. Locke, and J. S. Kearney, in addition to the officers.

Chapter members have always been active in the affairs of the Society and several past presidents reside in Chicago: Messrs. Lewis, Hill, Howatt, Hart. E. N. McDonnell is a member of the Council; M. K. Fahnestock is member of the Committee on Research. Last year E. P. Heckel was *chairman* of the ASHVE's Committee on Admission and Advancement, and P. D. Close served as *chairman* of the Guide Publication Committee.

The Illinois Chapter has always taken a keen interest in the Annual and Semi-Annual meetings of the national Society.

# The Resistance to Heat Flow Through Finned Tubing

By W. H. Carrier* and S. W. Anderson,** Syracuse, N. Y.

THE PAST twenty years have witnessed a great advance in the art of heating and cooling air. Prior to this period, prime surface was employed almost exclusively. Today, prime surface has been largely replaced by extended or finned surfaces. This advance in the art has been accelerated by the availability of non-ferrous metals of relatively low thermal resistance but at the same time highly resistant to external and internal corrosion. While technical advances have made this change practicable, the engineering and economic stimulus for the substitution of non-ferrous metals with extended surfaces has been largely provided by the growth of air conditioning with its more critical requirements for heat transfer surfaces—types that would be compact, efficient, and at the same time highly resistant to corrosion in handling moisture-laden air.

While surfaces of this character had long been successfully used in automobile radiators, it was not until a little over twenty years ago that this type of construction was successfully adapted to the art of heating, ventilating and air conditioning. It at once filled a great need in this art which was then at the very beginning of its period of rapid growth. Thus we see that the timing of this development was most auspicious both for the commercial development of such surfaces and for its indispensable assistance in the development and in the commercial exploitation of air conditioning.

From the foregoing it will be seen that the technique of design of finned heat transfer surfaces, particularly as employed in heating and ventilation, is of very considerable commercial importance as well as technical or engineering interest. These design factors are also of great practical importance in the design of fins for air-cooled airplane motors as well as in many other industrial applications.

In any heat transfer surface, there are three factors which determine the rate of heat exchange:

1. The resistance to heat flow through the film of the heating or cooling fluid to the conducting metal surface.
2. The resistance to heat flow of the metal itself from the inner surface to the outer surface.
3. The resistance to heat flow from the outer metal surface to the air stream which is to be heated or cooled.

In the use of prime surface in heating or cooling of air, the first resistance between fluid and wall is relatively small and the resistance of the metal wall itself is wholly negligible. With finned surface, however, the finned surface on the air side may be from 10 to 20 times that of the prime surface in contact with the heating or cooling fluid. Furthermore, the length of the path of heat flow through the metal itself is many times that through the prime surface. From this it may be seen that for a given rate of heat flow, resistances (1) and (2), which in previous practice were insignificant, now have an importance almost equalling that of the resistance on the air side. In fact, in many applications the temperature drop from surface to air is not more than one-half of the total temperature drop, while a temperature drop of more than about 70 per cent on the air side can hardly be considered the best economic practice.

For many years, manufacturers of finned heating and cooling surfaces have had data that was applicable in determining the performance of a particular design of finned surface for both the heating and cooling of air, including condensation of moisture. However, little data is available in usable form for calculating the effect of dimensional changes in the surface itself. It is the purpose of this paper to provide design data in a readily usable form which will facilitate optimum design for this type of heater for any service.

The principal design factors involved are *first*, the ratio of prime surface to extended surface, which in turn involves the spacing of the fin and the lineal extension of the fin beyond the prime surface. *Second*, the thickness of the fin and the conductivity of the metal employed.

Except in certain special applications where weight and space are at a premium, the *only* proper basis of comparison of two heat exchange designs is the relative cost of two designs that give identical per-

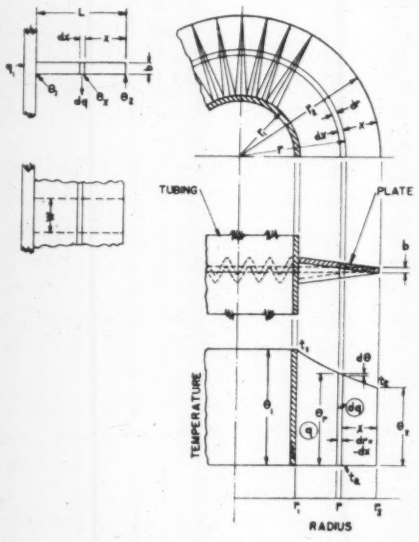


**Fig. 1—Bar fin and equivalent finned tube**

LEGEND

$b$ = Plate thickness, feet
$r$ = Radius, feet
$r_1$ = Inside radius of plate
= Outside radius of tubing
$r_2$ = Outside radius of plate
$dr$ = Differential radius
$x = r_2 - r$
$dx = -dr$
$L$ = Bar fin height, feet
$t$ = Temperature, degrees Fahrenheit
$t_1$ = Surface temperature at $r_1$
$t_2$ = Surface temperature at $r_2$
$t_a$ = Temperature of ambient fluid
$t_m$ = Mean surface temperature of plate
$W$ = $2\pi r_2$, feet
$\theta$ = Temperature difference, degrees Fahrenheit
$\theta_1 = t_1 - t_a$
$\theta_2 = t_2 - t_a$
$\theta_r = t_r - t_a$
$\theta_m = t_m - t_a$
$\theta_x = \theta_r$
$d\theta$ = Differential temperature difference
$Q$ = Quantity of heat, Btu
$q$ = Rate of heat flow, Btu per hour
$dq$ = Differential rate of heat flow

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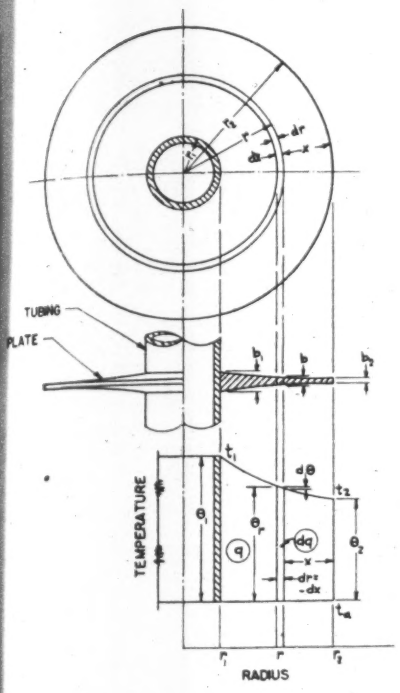


**Fig. 2—Constant metal cross section circular plate heating fluid diagram**

LEGEND

$b$ = Plate thickness, feet
$b_1$ = Plate thickness at $r_1$
$b_2$ = Plate thickness at $r_2$
$r$ = Radius, feet
$r_1$ = Inside radius of plate
= Outside radius of tubing
$r_2$ = Outside radius of plate
$dr$ = Differential radius
$x$ = $r_2 - r$
$dx$ = $-dr$
$t$ = Temperature, degrees Fahrenheit
$t_1$ = Surface temperature at $r_1$
$t_2$ = Surface temperature at $r_2$
$t_a$ = Temperature of ambient fluid
$t_m$ = Mean surface temperature of plate
$\theta$ = Temperature difference, degrees Fahrenheit
$\theta_1$ = $t_1 - t_a$
$\theta_2$ = $t_2 - t_a$
$\theta_r$ = $t_r - t_a$
$\theta_m$ = $t_m - t_a$
$d\theta$ = Differential temperature difference
$Q$ = Quantity of heat, Btu
$q$ = Rate of heat flow, Btu per hour
$dq$ = Differential rate of heat flow

formance. There are two criteria of performance which must be the same for both.

1. The two units must handle the *same quantity* of air at the *same pressure drop.* (The face area and face velocity may be quite different.)

2. The ratio of final to initial temperature difference must be identical.

The factor which is most difficult either to calculate or to determine experimentally is the average conductance of the fin itself. In many designs, particularly that of the flat disc or plate fin type of extended surface, the metal resistance is the determining factor in optimum design.

The principal contribution made by this paper is a temperature ratio chart, Fig. 5, which determines the resistance of any design of flat plate surface including (1) any diameter of tubing, (2) any spacing of tubing, and (3) any thickness or material of the fins.

The resistance to air flow and the corresponding air side heat transfer factor can be calculated independently to only an approximate accuracy. Final values for the performance in these two respects must be checked by tests. It is not the purpose of this paper to discuss this phase of the design as affecting the performance, but a correct method of comparison of two different types of heaters is developed and a formula for such comparison is given.

The general method of approach to the mathematical solution of a problem of metal resistance and the final results obtained by the application of the mathematical equations thus deduced will be given in a usable form in the body of the paper. The several mathematical solutions involved will be given in appropriate appendices for the convenience of those few who care to give the subject more critical study.

At least one of these solutions has previously been made and published by others.[1] Such derivations, however, have applied primarily to a bar fin of uniform thickness. This has a relatively simple mathematical solution while the mathematics of the plate fin transfer, to the authors' knowledge, has not been adequately covered as attempted in this paper. The material and calculations contained in this paper were made under the authors' direction some years ago and are now being taken out of the *mothballs.*

Commercial finned surfaces may be divided into three classifications, each of which has its own mathematical solution. In each of these forms, certain simplifying assumptions must be made in order to permit an accurate mathematical solution. In general, these assumptions are not widely at variance with practical results and are at least of comparative accuracy, which is the essential thing. The principal assumptions are:

1. That there is a known and fixed surface temperature at the tube.

2. That the coefficient of heat conduction from air to surface is constant for all points on the surface. This is only approximately true as the turbulence varies the conductance. Also, in condensing moisture from the air, the area nearest the tubing having a greater temperature differential gives an increased conductance in this area. The average value, however, is probably not far out of line although the actual performance should be somewhat better than that calculated from the chart.

3. That the lines of heat flow are approximately radial.

4. That the error introduced by the assumption of logarithmic mean temperature over the surface of the fin for any single row of tubes in depth as determining the actual temperature difference is negligible. (This would be physically correct if it were not for the fact that the variation in air temperature distorts the line of heat flow and lengthens the

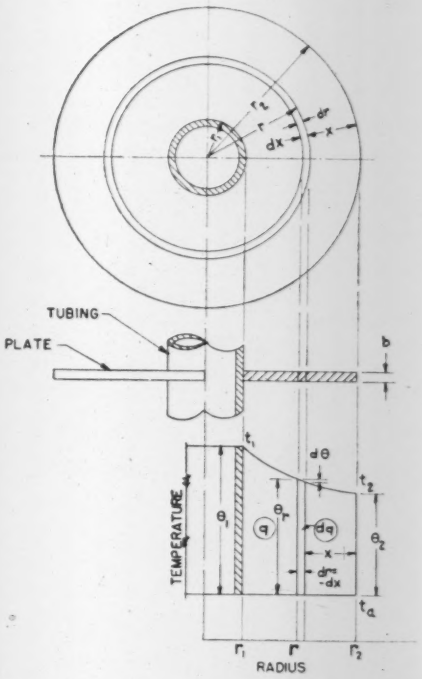


**Fig. 3—Flat circular plate heating fluid diagram**

LEGEND

$b$ = Plate thickness, feet
$r$ = Radius, feet
$r_1$ = Inside radius of plate
= Outside radius of tubing
$r_2$ = Outside radius of plate
$dr$ = Differential radius
$x$ = $r_2 - r$
$dx$ = $-dr$
$t$ = Temperature, degrees Fahrenheit
$t_1$ = Surface temperature at $r_1$
$t_2$ = Surface temperature at $r_2$
$t_a$ = Temperature of ambient fluid
$t_m$ = Mean surface temperature of plate
$\theta$ = Temperature difference, degrees Fahrenheit
$\theta_1$ = $t_1 - t_a$
$\theta_2$ = $t_2 - t_a$
$\theta_r$ = $t_r - t_a$
$\theta_m$ = $t_m - t_a$
$d\theta$ = Differential temperature difference
$Q$ = Quantity of heat, Btu
$q$ = Rate of heat flow, Btu per hour
$dq$ = Differential rate of heat flow

[1] S. R. Parsons and D. R. Harper, Nat. Bur. Standards Technol. Paper 211, p. 326, 1922; D. R. Harper and W. B. Brown, Nat. Adv. Comm. Aeronaut. Rept. 158, 1922, Government Printing Office, Washington, D. C.; E. Griffiths, Brit. Adv. Comm. Aero. Rept. and Memo. 308, 1917.

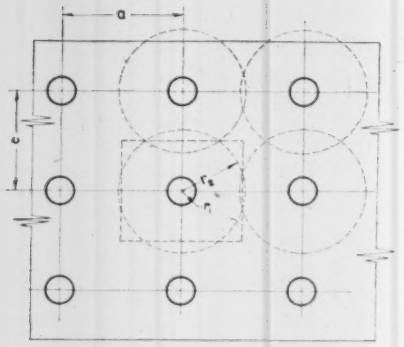

**Fig. 4—Section of three-row flat fin coil showing equivalent circular and rectangular heat transfer areas served by circular tubes**

path somewhat. Therefore, this assumption will give resistances somewhat below the actual.)

5. In plate fin construction, the assumption is made that the area served by each tube is equivalent in performance to a circular plate of equal area. The magnitude of the error introduced by this assumption either for a square or a rectangular spacing of tubes is discussed in Appendix IV, in which the approximate corrections to be applied are indicated.

The three main types of fins in commercial use are indicated in Figs. 1, 2 and 3. From the consideration of the geometry of each type of surface and the physical laws of heat transfer involved in conformance with the foregoing assumptions, a differential equation has been set up for each case and solved in the manner described in Appendices I, II and III. The first case, Fig. 1, corresponds to the solution for a *bar* fin. This fin may also be formed by winding a thin, narrow ribbon spirally about a tube. In this fin, the cross-section for transfer of heat is constant throughout the height of the fin and the perimeter is also constant throughout the height of the fin as the inner edge of the fin next to the tube is crimped but the metal is not appreciably stretched. In the second case, Fig. 2, a thicker ribbon is employed so that while the inner edge is not appreciably deformed, the outer edge is stretched and the metal thinned. In this fin, the cross-sectional area cut by a circle at any radius is constant but the perimeter or surface element exposed to air flow varies directly as the radius. In the third case, Fig. 3, which is the one primarily discussed in this paper, plates or discs of uniform thickness take the place of a spirally wound fin and the tubes pass perpendicularly through these plates of uniform thickness. In the latter case, cross-sectional area for heat transfer varies directly as the radius and the perimeter or differential area subject to air flow also varies directly as the radius. This is the case of the plate fin.

The differential equations for the three typical constructions are as follows:

$$\frac{d^2\Theta}{dx^2} = m^2\Theta \quad \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

$$\frac{d^2\Theta}{dx^2} = \frac{m^2\Theta}{r_1}(r_2 - x) \quad \ldots\ldots (2)$$

$$\frac{d^2\Theta}{dx^2} - \left(\frac{1}{r_2 - x}\right)\frac{d\Theta}{dx} = m^2\Theta. \quad (3)$$

In the Equations 1, 2 and 3 the following symbols are employed:

$x = (r_2 - r)$ = the distance from the outside edge to the radius $r$, feet

$r$ = the radius at any point in the fin to the center of the tube, feet

$r_1$ = outside radius of tube = inside radius of fin, feet

$r_2$ = outside radius of fin or equivalent, feet

$L$ = height of the fin, feet

$b$ = thickness of fin in feet

$\theta$ = the difference between the fin temperature at the radius $r$ and the ambient air temperature

$\theta_1$ = the temperature difference between the fin and the ambient air at the radius $r_1$

$\theta_2$ = difference between the temperature of the fin and the ambient air at radius $r_2$

$\theta_m$ = effective mean temperature difference between the fin and ambient air

$h$ = Btu per hour per square foot per degree difference = heat transfer coefficient from plate to air

$k$ = Btu per hour per square foot per degree temperature difference per foot = conductivity of finned metal

$E = \theta_m/\theta_1$ = fin effectiveness, dimensionless

The final solution for each of these cases (see Appendix I, II and III) is as follows:

Case I

$$\theta = \theta_2 \cdot \tfrac{1}{2}\,(e^{mx} + e^{-mx})$$

$$= \theta_1\left[\frac{e^{mx} + e^{-mx}}{e^{mL} + e^{-mL}}\right]$$

LEGEND

$P = r_1\sqrt{\frac{2h}{bk}}$ dimensionless quantity

$h$ = Fluid convection coefficient, Btu per hour, per square foot surface per degree Fahrenheit difference

$b$ = Plate thickness, feet

$k$ = Metal thermal conductivity, Btu per hour, per square foot per degree Fahrenheit difference per foot

$r_1$ = Inside radius of plate area = Outside radius of tubing, feet

$r_2$ = Outside radius of plate area, feet

$R$ = $r_2/r_1$ dimensionless quantity

$\theta_1$ = Maximum surface temperature difference, degree Fahrenheit

$\theta_m$ = Mean surface temperature difference, degree Fahrenheit

$E$ = $\theta_m/\theta_1$ dimensionless quantity

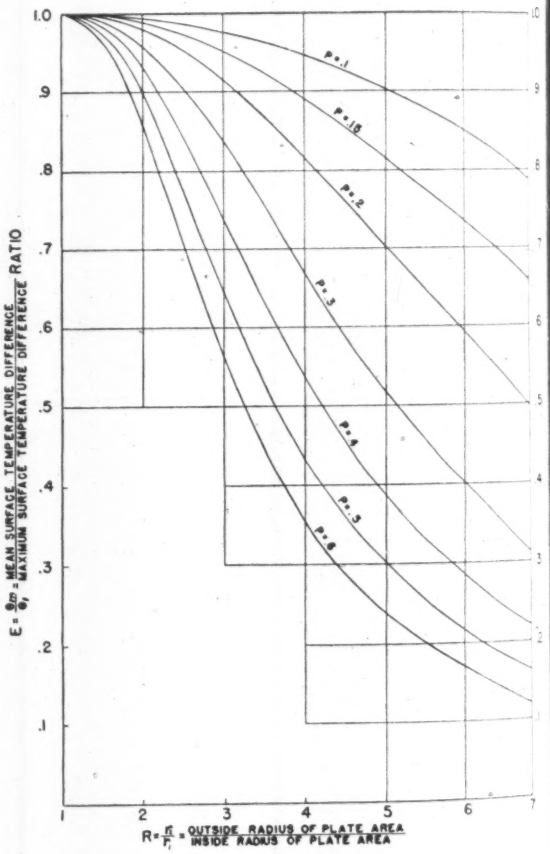


**Fig. 5—Curves showing heat transfer effectiveness from a flat circular or equivalent rectangular plate (*P* vs. *E* and *R*—residual included)**

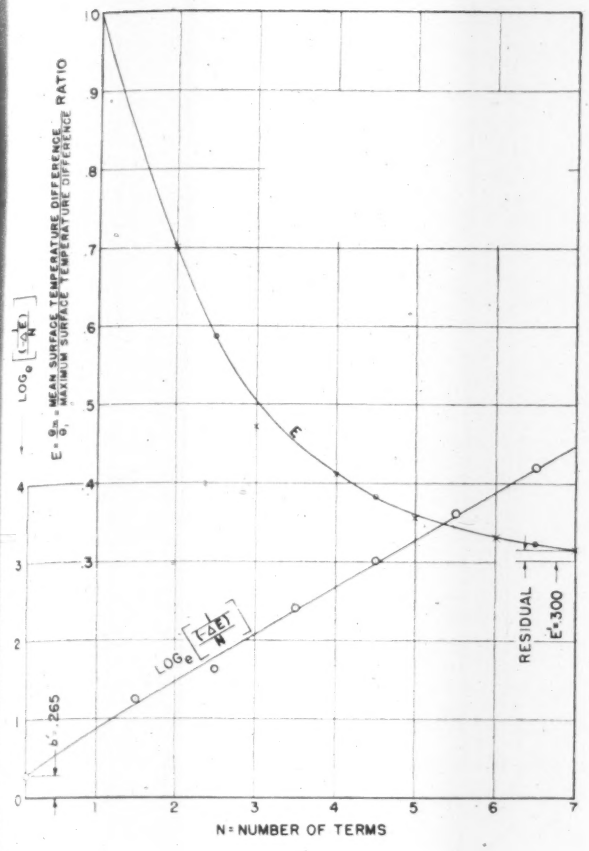


**Fig. 6—Curves showing variation of heat transfer effectiveness (*E* value) with number of terms (*N*) for point where $P = 0.5$ and $R = 5$ (Appendix III)**

Method of determining correction for residual error between seven terms and an infinite number of terms for Case III

$$\frac{\theta_m}{\theta_1} = \frac{\tanh\,[m\,(r_2 - r_1)]}{m\,(r_2 - r_1)}$$

where

$$m = \sqrt{\frac{2h}{kb}}$$

Case II

$$E = \frac{\theta_m}{\theta_1} = \frac{2R}{R+1}\left[\frac{1 - \frac{(R-1)}{2R} + P^2\left\{(R-1)^2\left(\frac{R+3}{24}\right) - \frac{(R-1)^2}{R}\left(\frac{R+4}{120}\right)\right\} + \ldots.}{1 + P^2\left\{\frac{R(R-1)^2}{2} - \frac{(R-1)^3}{6}\right\} + \ldots..}\right]$$

where

$R = r_2/r_1$

$P = r_1\sqrt{2h/kb_1}$

$b_1$ = fin metal thickness at fin base

Case III

$$E = \frac{\theta_m}{\theta_1} = \left[\frac{\frac{2R}{R+1}\left\{1 - \left(\frac{R-1}{2R}\right) + \frac{P^2}{6}(R-1) - \frac{P^2}{12R}(R-1)^3 - \frac{(P^2 - P^4R^2)}{120R^2}(R-1)^4 + \ldots..\right\}}{1 + \frac{P^2}{2}(R-1)^2 + \frac{P^2}{6R}(R-1)^3 + \left(\frac{P^2}{8R^2} + \frac{P^4}{24}\right)(R-1)^4 + \ldots..}\right]$$

where

$R = r_2/r_1$

$P = r_1\sqrt{2h/kb}$

Numerical values for Case I are easily found. Case II and Case III, however, can only be solved by a power series as indicated and complete numerical values have only been worked out for Case III. These have been plotted in Fig. 5, which permits the mean temperature to be calculated immediately when the size of spacing of the tubes, the thickness of the fin and the coefficient of heat transfer on the air side are known. It will be noted that this chart is plotted in terms of dimensionless groups, which makes it of universal application.

While the ratio of $\frac{\theta_m}{\theta_1}$ is shown to be a function of $h$, the actual resistance of the fin to the flow of heat varies but slightly with $h$ except for large values of $P$ and $\frac{r_2}{r_1}$. This is discussed and the variation shown in Appendix V. For this reason, for most practical cases, $R_m$ may be assumed to be a constant for different values of $h$ in the formula $R_m = \left[\frac{\theta_1}{\theta_m} - 1\right]\frac{1}{h}$, where $h$ is given an average value of 12 Btu per hour per square foot per degree difference. A chart, Fig. 5, based on this assumption for Case III permits the resistance of the plate fin to be calculated with sufficient accuracy within the commercial limitations regardless of the variation in $h$.

A few illustrations of the practical use of these formulae and charts are desirable.

## Comparison of Performance of Two Heaters Based on Test Results of Each

As mentioned in the forepart of the paper, the performance of two heaters can only be compared properly when they are tested or *calculated* for the same ratio of initial to final temperature difference and for the same pressure drop.[2] This relationship is developed in Appendix VI. The formula for this relationship is:

$$\frac{U_2}{U_1} = \left[\frac{\Delta p_2/\Delta p_1}{\log R_2/\log R_1}\right]^{m/(1+n-m)}$$

[2]Report No. 731—A Proposed Method for Comparison of Effectiveness of Indirect Heating Surfaces, by A. E. Stacey, Jr., and C. M. Ashley. (ASHVE Transactions, Vol. 31, 1925, p. 257.)

## Table 1—Bar Fin Equivalent (Case I)

| Example No. | | A | B |
|---|---|---|---|
| | Surface material | Copper | Mild Steel |
| | Fin spacing | 8/inch | 8/inch |
| | Tube arrangement in adjacent rows | Staggered | Staggered |
| | Rows deep | 4 | 4 |
| $a$ = | Tube face spacing | 0.116ft | 0.116 ft |
| $c$ = | Tube row spacing | 0.100 ft | 0.100 ft |
| $b$ = | Fin thickness | 0.000833 ft | 0.00125 ft |
| $r_2$ = | Outside fin radius | 0.0580 ft | 0.0580 ft |
| $r_1$ = | Fin base radius | 0.0260 ft | 0.0260 ft |
| $L$ = | $r_2 - r_1$ = Fin height | 0.0320 ft | 0.0320 ft |
| $r_1'$ = | Inside tube radius | 0.0237 ft | 0.0237 ft |
| $h_1$ = | Fluid convection coefficient inside tube | 600 | 600 |
| $h_a$ = | Air convection coefficient between fin and ambient air | 12 | 12 |
| $k$ = | Fin metal thermal conductivity | 220 | 30 |
| $m$ = | $\sqrt{2h_a/kb}$ | 11.45 | 25.3 |
| $mL$ = | | 0.366 | 0.810 |
| $E$ = | $\frac{\tanh mL}{mL} = \frac{\theta_m}{\theta_1}$ | 0.959 | 0.827 |
| $S/A$ = | Total exposed out. surf., sq ft surf./(sq ft face area) (4 rows deep) | 81.8 | 81.4 |
| $S_1/A$ = | Inside tube surf., sq ft surf./(sq ft face area) (4 rows deep) | 5.1 | 5.1 |
| $S_p/A$ = | Outside exposed tube surf., sq ft surf./(sq ft face area) (4 rows deep) | 4.6 | 4.1 |
| $S_f/A$ = | Lateral fin surf., sq ft surf./(sq ft face area) (4 rows deep) | 77.2 | 77.3 |
| $100\ (A_f/A)$ = | Per cent minimum free face area | 48.5 | 45.5 |
| $R_m$ = | $\frac{1}{h_a}\left[\frac{1-E}{E+\frac{S_p}{S_f}}\right]$ = Metal thermal resistance, (hr) (sq ft of $S$) (deg F)/Btu | 0.0034 | 0.0164 |
| $R_a$ = | $\frac{1}{h_a}$ = Air film thermal resistance, (hr) (sq ft of $S$) (deg F)/Btu | 0.0833 | 0.0833 |
| $R_1$ = | $\left(\frac{S}{S_1}\right)\frac{1}{h_1}$ = Inside tube film thermal resistance, (hr) (sq ft of $S$) (deg F)/Btu. | 0.0265 | 0.0264 |
| $R_S$ = | $R_a + R_m + R_1$ = Over-all thermal resistance, (hr) (sq ft of $S$) (deg F over-all)/Btu. | 0.1132 | 0.1261 |
| $U$ = | $1/R_S$ = Over-all heat transfer coefficient, Btu/(hr) (sq ft of $S$) (deg F over-all) | 8.83 | 7.92 |

## Table 2—Heater Comparison—Bar Fin Equivalent (Case I)

*(Copper vs. Steel Fin Heater Performance)*

| Example No. | | A | B | BC |
|---|---|---|---|---|
| $V$ = | Air face velocity, ft min | 553 | 520 | .... |
| $S/A$ = | Total outside surface, sq ft surf./(sq ft face) (4 rows deep) | 81.8 | 81.4 | .... |
| $G$ = | Air mass velocity, lb/(hr) (sq ft face) | 2490 | 2340 | .... |
| $U$ = | Over-all heat transfer coefficient Btu/(hr) (sq ft of $S$) (deg F over-all) | 8.83 | 7.92 | .... |
| $C_p$ = | Air specific heat, Btu/(lb) (deg F) | 0.24 | 0.24 | .... |
| $m$ = | Air velocity heat transfer exponent | 0.51 | 0.46 | .... |
| $n$ = | Air velocity exponent of air friction | 1.81 | 1.81 | .... |
| $\log R$ = | $\frac{U(S/A)}{C_pG}$ = Log of temperature difference ratio, $(\theta_1/\theta_2)$ | 1.21 | 1.15 | 1.21 |
| $\Delta p$ = | Air friction/4 rows deep (inches of water) | 0.51 | 0.51 | 0.51 |
| $U_{BC}$ = | $U_B\left[\frac{\Delta p_{BC}/\Delta p_B}{\log R_{BC}/\log R_B}\right]^{(m/1+n-m)}$ | .... | .... | 7.83 |
| $U_{BC}/U_A$ = | $\frac{7.83}{8.83}$ | .... | .... | 0.887 |

## Table 3—Flat, Plate Fin (Case III)

| Example No. | | A | B |
|---|---|---|---|
| | Surface material | Copper | Copper |
| | Tube arrangement in adjacent rows | *In Line* | *In Line* |
| | Fin spacing | 6/in. | 6/in. |
| | Rows deep | 4 | 4 |
| $a$ = | Tube face spacing = Fin length/tube | 0.166 ft | 0.208 ft |
| $c$ = | Tube row spacing = Fin depth/tube | 0.166 ft | 0.125 ft |
| $b$ = | Fin thickness | 0.000833 ft | 0.000833 ft |
| $r_2$ = | Radius of equivalent circular area = $\sqrt{\frac{ac}{\pi}}$ | 0.0940 ft | 0.0911 ft |
| $r_1$ = | Fin base radius | 0.0260 ft | 0.0260 ft |
| $r_1'$ = | Inside tube radius | 0.0237 ft | 0.0237 ft |
| $h_1$ = | Fluid convection coefficient inside tube | 600 | 600 |
| $h_a$ = | Air convection coefficient between fin and ambient air | 10 | 10 |
| $k$ = | Fin metal thermal conductivity | 220 | 220 |
| $r_2/r_1$ = | See Appendix IV, *Applications* | 3.69 | 3.58 |
| $P$ = | See Appendix IV, *Applications* | 0.272 | 0.272 |
| $\overline{E}$ = | $\theta_m/\theta_1$ See Appendix IV, *Applications* | 0.768 | 0.772 |
| $S/A$ = | Total exposed outside surf., sq ft surf./(sq ft face area) (4 rows deep) | 92.5 | 69.0 |
| $S_1/A$ = | Inside tube surf., sq ft surf./(sq ft face area) (4 rows deep) | 3.7 | 2.9 |
| $S_p/A$ = | Outside exposed tube surf., sq ft surf./(sq ft face area) (4 rows deep) | 3.7 | 2.9 |
| $S_f/A$ = | Lateral fin surf., sq ft surf./(sq ft face area) (4 rows deep) | 88.8 | 66.1 |
| $100\ (A_f/A)$ = | Per cent minimum free face area | 64.5 | 70.5 |
| $R_m$ = | $\frac{1}{ha}\left[\frac{1-\overline{E}}{\overline{E}+\frac{S_p}{S_f}}\right]$ = Metal thermal resistance, (hr) (sq ft of $S$) (deg F)/Btu | 0.0287 | 0.0280 |
| $R_a$ = | $1/h_a$ = Air film thermal resistance, (hr) (sq ft of $S$) (deg F)/Btu | 0.1000 | 0.100 |
| $R_1$ = | $\left(\frac{S}{S_1}\right)1/h_1$ = Inside tube film thermal resistance, (hr) (sq ft of $S$) (deg F)/Btu. | 0.0430 | 0.040 |
| $R_S$ = | $R_a + R_m + R_1$ = Over-all thermal resistance, (hr) (sq ft of $S$) (deg F over-all)/Btu | 0.1717 | 0.168 |
| $U$ = | $1/R_S$ = Over-all heat transfer coefficient, Btu/(hr) (sq ft of $S$) (deg F over-all) | 5.82 | 5.95 |

## Table 4—Heater Comparison—Flat, Plate Fin (Case III)

*(Square vs. Rectangular Plate Fin Heater Performance)*

| Example No. | | A | B | BC |
|---|---|---|---|---|
| $V$ = | Air face velocity, ft min | 1250 | 1370 | .... |
| $S/A$ = | Total outside surface, sq ft surf./(sq ft face) (4 rows deep) | 92.5 | 69.0 | .... |
| $G$ = | Air mass velocity, lb/(hr) (sq ft face) | 5630 | 6170 | .... |
| $U$ = | Over-all heat transfer coefficient Btu/(hr) (sq ft of $S$) (deg F over-all) | 5.82 | 5.95 | .... |
| $C_p$ = | Air specific heat, Btu/(lb) (deg F) | 0.24 | 0.24 | .... |
| $m$ = | Air velocity heat transfer exponent | 0.38 | 0.38 | .... |
| $n$ = | Air velocity exponent of air friction | 1.71 | 1.71 | .... |
| $\log R$ = | $\frac{U(S/A)}{C_pG}$ = Log of temperature difference ratio, $(\theta_1/\theta_2)$ | 0.398 | 0.277 | 0.398 |
| $\Delta p$ = | Air friction/4 rows deep (inches of water) | 0.54 | 0.45 | 0.54 |
| $U_{BC}$ = | $U_B\left[\frac{\Delta p_{BC}/\Delta p_B}{\log R_{BC}/\log R_B}\right]^{(m/1+n-m)}$ | 5.78 | .... | .... |
| $U_{BC}/U_A$ = | $\frac{5.78}{5.82}$ | .... | .... | 0.99 |

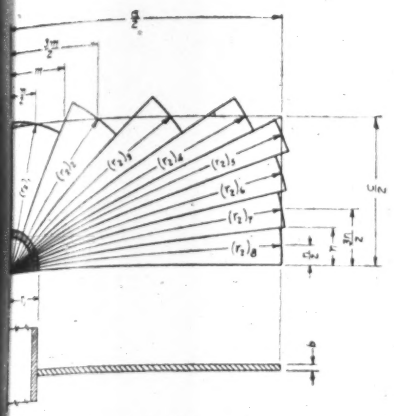

**Fig. 7—Rectangular plate fin quadrant showing segmentation (Appendix IV)**

LEGEND

$a$ = Fin plate length per tube, feet
$b$ = Fin plate thickness, feet
$c$ = Fin plate width per tube, feet
$r_2$ = Outside sector radius, feet
$r_1$ = Outside tube radius, feet
$N$ = Total number of sectors
$m = a/N$ = fin plate length segment, feet
$n = c/N$ = fin plate width segment, feet

where $U_1$, $p_1$, and $R_1$ are taken from a test point on an actual heater. $\triangle p_2$ and $R_2$ are assumed conditions to be met by the same heater with required change in the heater of the ratio of depth to face area and $U_2$ is the corresponding heat transfer value under these conditions. m and n are air velocity exponents expressing both $u$ and $\triangle p$, respectively, in an exponential relationship with the heater face area velocity.

## Illustrations of Practical Application of Formulae and Charts

A few illustrations of the practical use of these formulae and charts is desirable.

Assume A a copper tube and fin and B a tube and fin of mild steel of the specifications given in the preceding tabulation. The relative results are also given in the same tabulation. Tabulations for these examples appear in Tables 1 and 2.

It will be noted from this tabulation that the overall heat transfer is 8.83 for the copper and 7.92 for the steel. However, while the pressure drop for both heaters is the same, the air face velocity is different. The ratio of initial to final temperature difference is different for the two cases. Therefore, we will have to make a comparison applying the formula of Appendix VI for comparing two different heater surfaces. This is done in the preceding tabulation, in which we see that the relative value of the steel surface is approximately 89 per cent of that of the copper surface. If the cost of the steel surface as compared to the cost of the copper surface was approximately in the same ratio, there would be no choice between steel and copper commercially, aside from difference in resistance to corrosion. As a matter of fact, however, the steel surface actually costs somewhat more to produce than the copper surface, so there is no question when postwar restrictions are removed that copper will be used in preference to steel.

Let us now consider two examples under Case III, one A for a plate fin with square spacing of tubes and the other B for a rectangular spacing of tubes. On the basis of a comparison of the two performances, which are somewhat different, by the formula of Appendix VI it will be seen that the ratio of heat transfer under identical conditions of B is 99 per cent of that of A or, in other words, substantially the same, being an advantage of only 1 per cent of square arrangement over rectangular arrangement in the particular examples given. These examples are shown in Tables 3 and 4.


### Acknowledgment

The authors wish to acknowledge the assistance of Robert B. Hyde in making original numerical computations for the flat plate performance chart, and to Byron W. Winborn, Jr., for suggesting the method of computing corrections for rectangular fins compared with equivalent circular fins of equal area, Appendix IV.


## Appendix I

### Derivation of the Formula for Transfer and Temperature Drop Through the Bar Fin or Equivalent Spiral Finned Tube—Case I

This is the simplest pattern of heat distribution in finned surfaces and has been amply covered by others.[3] The derivation of its mathematics is given here because of its relation to Cases II and III. The following symbols which refer to Fig. 1 are employed:

$L$ = height of the fin—feet

$x$ = distance from outer edge of the fin to an element $dx$

$dq = -h\theta dS$ = differential change in rate of heat flow through the laminar section $dx \times bW$ which is located at the distance $x$ from the outer edge of the fin

$b$ = thickness of fin—feet

$W = (2\pi r_2)$ = width of any fixed segment of the fin—feet

$dS = -2Wdx$ = differential area from which heat is transferred to the ambient air—square feet

$q$ = rate of heat flow through any laminar section of the fin at the distance $x$ from the outer edge—Btu per hour

$q_1$ = rate of heat flow through a section at the base of the fin

$q_2 = O$ (for convenience of calculation, it is assumed there is no heat conducted from the edge of the fin)

$\theta$ = temperature difference between the ambient air and the fin at the distance $x$ from the outer edge of the fin

$\theta_1$ = temperature difference between the air and the base of the fin

$\theta_2$ = temperature difference between the ambient air and the outer edge of the fin

$h$ = coefficient of heat transfer from a surface to the air

$k$ = thermal conductivity of the fin metal

The law of temperature change in this surface (or any other) may be expressed by a differential equation formed by equating two expressions for change in heat flow; *viz*, (1) the rate of heat flow from the differential area $Wdx$ to the air stream, $dq = h\theta\ (2Wdx)$, and (2) the rate of heat flow through the cross sectional area $Wb$ as derived from the resistance formula $\frac{d\theta}{dx} = \frac{q}{kWb}$.

[3] S. R. Parsons and D. R. Harper, Nat. Bur. Standards Technol. Paper 211, p. 326, 1922.

**Then:**

$$\frac{dq}{dx} = 2h\theta W \text{ from the first} \quad \ldots\ldots (1)$$

$$\frac{d\theta}{dx} = \frac{q}{kWb} \text{ from the second} \quad \ldots\ldots (2)$$

Taking the derivative of both sides of Equation (2) with respect to $x$, we have

$$\frac{d^2\theta}{dx^2} = \frac{1}{kWb}\frac{dq}{dx} \text{ or} \quad \ldots\ldots (3)$$

$$\frac{dq}{dx} = kWb\frac{d^2\theta}{dx^2} \quad \ldots\ldots (4)$$

Combining Equations (1) and (4),

$$\frac{d^2\theta}{dx^2} = \left(\frac{2h}{kb}\right)\theta \text{ or } \frac{d}{dx}\left(\frac{d\theta}{dx}\right) = \left(\frac{2h}{kb}\right)\theta \quad \ldots (5)$$

This basic differential equation may be solved very simply by ordinary integration as follows: Multiply both sides of Equation (5) by $\left(\frac{d\theta}{dx}\right)$ Then

$$\frac{d\theta}{dx}\cdot\frac{d}{dx}\left(\frac{d\theta}{dx}\right) = \left(\frac{2h}{kb}\right)\theta\frac{d\theta}{dx} \quad \ldots\ldots (6)$$

This is equivalent to

$$\frac{1}{2}\frac{d}{dx}\left(\frac{d\theta}{dx}\right)^2 = \left(\frac{2h}{kb}\right)\theta\frac{d\theta}{dx} \text{ or}$$

$$\frac{1}{2}d\left(\frac{d\theta}{dx}\right)^2 = f(x)\,dx = \left(\frac{2h}{kb}\right)\theta d\theta \quad \ldots (7)$$

Integrating the first term of Equation (7) between the limits $x = O$ and $x = x$ and corresponding limits $\frac{d\theta}{dx} = O'$ and $\frac{d\theta}{dx} = \frac{d\theta}{dx}$, and integrating the second member of the second equation between the corresponding limits $\theta_2$ and $\theta$, we have

$$\frac{1}{2}\left(\frac{d\theta}{dx}\right)^2 - \left(\frac{1}{2}\times O\right) = \frac{1}{2}\left(\frac{2h}{kb}\right)[\theta^2 - \theta_2^2] \text{ or}$$

$$\frac{d\theta}{dx} = \sqrt{\frac{2h}{kb}}\sqrt{\theta^2 - \theta_2^2} \text{ or}$$

$$\sqrt{\frac{2h}{kb}}\,dx = \frac{d\theta}{\sqrt{\theta^2 - \theta_2^2}} \quad \ldots\ldots (8)$$

Then

$$\sqrt{\frac{2h}{kb}}\int_o^x dx = \int_{\theta_2}^{\theta}\frac{d\theta}{\sqrt{\theta^2 - \theta_2^2}} \text{ or}$$

$$x\sqrt{\frac{2h}{kb}} = \log\left[\frac{\theta + \sqrt{\theta^2 - \theta_2^2}}{\theta_2 + \sqrt{\theta_2^2 - \theta_2^2}}\right]$$

$$= \log\left[\left(\frac{\theta}{\theta_2}\right) + \sqrt{\left(\frac{\theta}{\theta_2}\right)^2 - 1}\right] = \cosh^{-1}\left(\frac{\theta}{\theta_2}\right) \quad \ldots (9)$$

This will give the temperature at any point on the fin if $\theta_2$ is known, including the point $x = L = (r_2 - r_1)$; *i.e.*, where $\theta$ equals $\theta_1$ which is known. Therefore,

$$L\sqrt{\frac{2h}{kb}} = \log\left[\frac{\theta_1}{\theta_2} + \sqrt{\left(\frac{\theta_1}{\theta_2}\right)^2 - 1}\right] \quad \ldots (10)$$

From Equation (10) and putting $\sqrt{\frac{2h}{kb}} = m$ for convenience

$$\frac{\theta_1}{\theta_2} = \frac{1}{2}\left[e^{mL} + e^{-mL}\right] = \cosh\left[L\sqrt{\frac{2h}{kb}}\right]$$

$$\theta_2 = \frac{\theta_1}{\frac{1}{2}(e^{mL} + e^{-mL})} \quad \ldots\ldots (11)$$

From Equations (9) and (11)

$$\theta = \theta_2\cdot\frac{1}{2}[e^{mx} + e^{-mx}] = \theta_1\left[\frac{e^{mx} + e^{-mx}}{e^{mL} + e^{-mL}}\right] \quad \ldots (12)$$

One is now able to integrate the rate of heat transfer to the air over the entire surface and also to determine the mean temperature difference, $\theta_m$. Then

$$\int_o^{q_s} dq = \int_o^{S_s} h\theta dS = \int_o^{L} 2hW\theta dx = \int_{r_1}^{r_2} 2hW\theta dr$$

See Equation (1).

Introducing the value of $\theta$ from Equation (12)

$$\int_o^{q_s} dq = 2hW\theta_2\int_o^{L}\frac{1}{2}[e^{mx} + e^{-mx}]\,dx \quad \ldots\ldots (13)$$

$$\text{Since } \int e^{mx}\,dx = \frac{1}{m}\int e^{mx}\,d(mx) = \frac{1}{m}e^{mx}$$

$$q_s = \frac{2hW\theta_2}{m}\cdot\frac{1}{2}[(e^{mL} - 1) - (e^{-mL} - 1)]$$

$$= \frac{2hW\theta_1}{m}\left[\frac{e^{mL} - e^{-mL}}{e^{mL} + e^{-mL}}\right] = 2hWL\theta_1\left[\frac{\tanh\ mL}{mL}\right] \quad \ldots (14)$$

noting that $mL$ is a dimensionless number.

$q_s = h\theta_m S_s$ where $S_s = 2WL$. Therefore,

$$\theta_m = \frac{q_s}{2hWL} = \frac{\theta_1}{mL}\left[\frac{e^{mL} - e^{-mL}}{e^{mL} + e^{-mL}}\right] \quad \ldots\ldots (15)$$

or

$$\frac{\theta_m}{\theta_1} = \frac{\tanh mL}{mL} \text{ where } m = \sqrt{\frac{2h}{kb}} \quad \ldots\ldots (15a)$$

or

$$\frac{\theta_m}{\theta_1} = \frac{\tanh m(r_2 - r_1)}{m(r_2 - r_1)} \quad \ldots\ldots (15b)$$

It will be seen that Equations (14) and (15) give the practical solutions required and a chart in terms of the dimensionless groups, $\frac{\theta_m}{\theta_1}$, $\frac{r_2}{r_1}$ and $P = r_1\sqrt{\frac{2h}{kb}}$ could be plotted as in Fig. 5, but this is not necessary as in Cases II and III because of the simplicity of the Case I solution.

## Resistance

The only remaining equation required is that giving the thermal resistance to heat flow through the fin in order to determine the overall conductance, $U = \frac{1}{R_s}$, where $R_s$ = the summation of three principal resistances.

[4] Since $q = O$ when $x = O$ and therefore $d\left[\frac{d\theta}{dx}\right] = \frac{q}{kWb} = O$ when $x = O$.

$$R_s = R_f + R_m + R_a = \frac{1}{h_f} + \frac{1}{C_m} + \frac{1}{h_a} \text{ where}$$

$R_f$ = the film resistance of the heating fluid on the inside of the tube referred to surface $(S)$[3]

$R_m$ = the metal resistance of the fin

$R_a$ = the resistance of the air film

and $h_f$, $C_m$, $h_a$ are the respective conductances referred to surface $(S)$[3].

In heat transfer under the steady state, the equation

$$q/S = C\ (t_1 - t_m) \text{ i.e. } C = \frac{d(q/S)}{d\theta}$$

$$\frac{(\theta_1 - \theta_m)}{R_m} = q/S = h\theta_m \text{ where } h \text{ is the coefficient of heat transfer from surface to air} \quad \ldots\ldots (16)$$

$$R_m = \frac{\theta_1 - \theta_m}{h\theta_m} = \frac{1}{h}\left[\left(\frac{\theta_1}{\theta_m}\right) - 1\right]$$

$$= \frac{1}{h}\left[\frac{1 - \left(\frac{\theta_m}{\theta_1}\right)}{(\theta_m/\theta_1)}\right] \quad \ldots\ldots (17)$$

This relationship obviously applies equally to Cases I, II and III or any others.

In Case I,

$$R_m = \frac{1}{h}\left[\frac{mL}{\tanh mL} - 1\right] \text{where } m = \sqrt{\frac{2h}{kb}} \quad \ldots (18)$$

Apparently this resistance is not constant, as one might suppose, but is an involved function of $h$. However, in most practical designs of finned tubing, the effect of the variation of $h$ through considerable range is nearly negligible, so that *the fin resistance $R_m$, calculated for an average value of $h$ is satisfactory for other values of $h$.* That is, it is not important that we know the actual value of $h$ in order to calculate the resistance of the fin.

## Appendix II

### Derivation of Formula for the Temperature Change Through a Circular Fin of Constant Metal Cross Section—Case II

*Symbols:*

As in Appendix I with the following exceptions:

$b_1$ = fin metal thickness at fin base, feet, since $b$ is variable for this case.

$R = \frac{r_2}{r_1}$, dimensionless

$m = \sqrt{\frac{2h}{kb_1}}, \frac{1}{\text{ft}}$

$P = r_1 m = r_1\sqrt{\frac{2h}{kb_1}}$, dimensionless

$E = \frac{\theta_m}{\theta_1}$ = fin effectiveness, dimensionless

### Development of Formula

The differential equation, expressing the temperature change through a fin of uniform metal cross section such as shown in Fig. 2, is determined in a manner similar to the detailed derivation procedure outlined in Appendix I.

The constant fin metal cross section requires that

$$r_1 b_1 = r_2 b_2 = rb$$

If

$$m = \sqrt{\frac{2h}{kb_1}}$$

and heat transmission through the outside fin edge area $(2\pi r_2 b_2)$ be neglected, the following differential equation is obtained:

$$\frac{d^2\theta}{dx^2} = \frac{m^2\theta\ (r_2 - x)}{r_1} \quad \ldots\ldots (1)$$

Equation (1), in Appendix II, is that of Case II in the text, and may be solved by expressing $\theta$ in terms of a power series of $x$. The procedure outlined below is employed to develop an expression for the fin effectiveness $E$.

The boundary conditions, from an inspection of Fig. 2, are as follows:

$\theta = \theta_2$, when $x = O$ ..... (a)

$\frac{d\theta}{dx} = O$, when $x = O$ (neglecting outside fin edge heat transmission) ..... (b)

$\theta = \theta_1$, when $x = (r_2 - r_1)$ ..... (c)

Let

$$\theta = A + Bx + Cx^2 + Dx^3 + \quad \ldots\ldots (2)$$

The first derivative of Equation (2) is

$$\frac{d\theta}{dx} = B + 2Cx + 3Dx^2 + \quad \ldots\ldots (3)$$

and the second derivative is

$$\frac{d^2\theta}{dx^2} = 2C + 6Dx + \quad \ldots\ldots (4)$$

Substitution of boundary condition $(a)$ in Equation (2) determines the coefficient $A$, thus

$$A = \theta_2.$$

Substituting boundary condition $(b)$ in Equation (3) establishes the coefficient $B$, thus

$$B = O.$$

The remaining undetermined coefficients are evaluated by substituting the values of $\theta$ and $\frac{d^2\theta}{dx^2}$ from Equations (2) and (4), respectively, in Equation (1), hence

$$2C + 6Dx + \ldots = \frac{m^2}{r_1}$$

$$[Ar_2 + (Br_2 - A)\ x + (Cr_2 - B)\ x^2 + \ldots] \quad \ldots\ldots (5)$$

Solving for coefficients in terms of the previously established values of $A$ and $B$, by equating coefficients of the like exponents of $x$ in Equation (5), we obtain

$$C = \frac{\theta_2 m^2 r_2}{2r_1}$$

$$D = -\frac{\theta_2 m^3}{6r_1} \quad \ldots\ldots \text{ etc.}$$

In a similar manner, any number of coefficients for the higher powers of $x$ are evaluated.

Substituting the determined coefficients in Equation (2), we find

$$\theta = \theta_2\left[1 + \left(\frac{m^2 r_2}{2r_1}\right)x^2 - \left(\frac{m^2}{6r_1}\right)x^3 + \ldots\right] \quad \ldots\ldots (6)$$

The total heat transmitted from the lateral fin surface to the ambient fluid, neglecting the outside fin edge, may be expressed as follows:

$$q = 2\pi(r_2^2 - r_1^2)\ h\theta_m = 4\pi h\int_0^{r_2 - r_1} \theta\ (r_2 - x)\ dx \quad \ldots (7)$$

Then from Equation (7),

$$\theta_m = \frac{2}{r_2^2 - r_1^2}\int_0^{r_2 - r_1} \theta\ (r_2 - x)\ dx \quad \ldots\ldots (7a)$$

Substituting the value of $\theta$ from Equation (6) in (7a)

$$\frac{\theta_m}{\theta_2} = \frac{2}{r_2^2 - r_1^2}\int_0^{r_2 - r_1}\left[r_2 - x + \frac{m^2 r_2}{2r_1}(r_2 x^2 - x^3)\right.$$

[3] All thermal resistances must be referred to the same surface. For example, the effective inside film resistance $(R_f)$ referred to the total outside surface $(S)$ is equal to the specific inside film resistance multiplied by the ratio of total outside surface to inside surface.

$$-\frac{m^2}{6r_1}(r_2x^3 - x^4) + \ldots\ldots\Big]dx \quad \ldots\ldots (8)$$

Integrating Equation (8), and substituting limits

$$\frac{\theta_m}{\theta_2} = \frac{2}{r_2 + r_1}\Bigg[r_2 - \frac{(r_2 - r_1)}{2} + \frac{m^2 r_2}{2r_1}$$

$$\frac{r_2(r_2 - r_1)^2}{3} - \frac{(r_2 - r_1)^3}{4}\Bigg\} - \frac{m^2}{6r_1}$$

$$\Bigg\{\frac{r_2(r_2 - r_1)^3}{4} - \frac{(r_2 - r_1)^4}{5} + \ldots\ldots\Bigg] \quad \ldots\ldots (9)$$

Let

$$r_2 = \frac{PR}{m} \quad \ldots\ldots (10)$$

and

$$r_1 = \frac{P}{m} \quad \ldots\ldots (11)$$

*where*

$$P = r_1\sqrt{\frac{2h}{kb_1}}$$

Substituting the values of $r_2$ and $r_1$ from Equations (10) and (11), respectively, in Equation (9)

$$\frac{\theta_m}{\theta_2} = \frac{2R}{R+1}\Bigg[1 - \frac{(R-1)}{2R} + P^2\Bigg\{(R-1)^2\left(\frac{R+3}{24}\right)$$

$$-\frac{(R-1)^3}{R}\left(\frac{R+4}{120}\right)\Bigg\} + \ldots\ldots\Bigg] \quad \ldots\ldots (12)$$

Substituting boundary condition ($c$) and the values of $r_2$ and $r_1$ from Equations (10) and (11), respectively, in Equation (6), we find the following relation

$$\frac{\theta_1}{\theta_2} = \Bigg[1 + P^2\,\frac{R(R-1)^2}{2} - \frac{(R-1)^3}{6}\Bigg\}$$

$$+ \ldots\ldots \quad \ldots\ldots (13)$$

The total heat transmission ($q$), from the lateral fin surface ($S_f$), is expressed as

$$q = hS_f\theta_m \quad \ldots\ldots (14)$$

or

$$q = hS_fE\theta_1 \quad \ldots\ldots (14a)$$

Equating the values of ($q$) from Equations (14) and (14$a$), the value of ($E$) in terms of the temperature difference ratio is

$$E = \frac{\theta_m}{\theta_1} \quad \ldots\ldots (15)$$

From Equation (15) it is obvious that the fin effectiveness ($E$), numerically equals that fraction of the lateral fin surface ($S_f$), which, if operated at the prime or tube surface temperature difference ($\theta_1$), would dissipate an amount of heat equaling the fin's total heat transmission ($q$).

Finally, the formula for the fin effectiveness ($E$), expressed as a function of the two dimensionless parameters, $R$ and $P$, is obtained by dividing Equation (12) by (13), hence

$$E = \frac{\theta_m}{\theta_1} = \frac{2R}{R+1}\left[\frac{1 + f_1(R) + P^2f_2(R) + \ldots.}{1 + P^2f_s(R) + \ldots.}\right] \quad (16)$$

Where

$$f_1(R) = -\left(\frac{R-1}{2R}\right)$$

$$f_2(R) = \frac{(R-1)^2}{24}\left[R + 3 - \frac{(R-1)(R+4)}{5R}\right]$$

$$f_s(R) = (R-1)^2\left[\frac{2R+1}{6}\right]$$

## Appendix III

## Derivation of Formula for the Temperature Change Through a Circular Fin Plate of Uniform Thickness—Case III

*Symbols*

The same as in Appendix I with the following exceptions:

$R = \frac{r_2}{r_1}$, dimensionless

$P = r_1m = r_1\sqrt{\frac{2h}{kb}}$, dimensionless

$E = \frac{\theta_m}{\theta_1}$ = fin effectiveness, dimensionless

### Development of Formula

The differential equation, expressing the temperature change through a circular fin of uniform thickness ($b$), such as illustrated in Fig. 3, is found in a manner similar to the detailed derivation procedure outlined in Appendix I.

If

$$m = \sqrt{\frac{2h}{kb}}$$

and heat transmission through the outside fin edge area ($2\pi r_2b$) be neglected, the following differential equation is obtained:

$$\frac{d^2\theta}{dx^2} - \left(\frac{1}{r_2 - x}\right)\frac{d\theta}{dx} = m^2\theta \quad \ldots\ldots (1)$$

Equation (1), above, is that of Case III in the text, and may be solved by expressing $\theta$ in terms of a power series of $x$. The forthcoming procedure is used to develop an expression for the fin effectiveness ($E$).

From an inspection of Fig. 3, the boundary conditions are established as follows:

$\theta = \theta_2$, when $x = O$ ..... ($a$)

$\frac{d\theta}{dx} = O$, when $x = O$ (neglecting outside fin edge heat transmission) ..... ($b$)

$\theta = \theta_1$, when $x = (r_2 - r_1)$ ..... ($c$)

Let

$$\theta = A + Bx + Cx^2 + Dx^3 + Ex^4 + \ldots\ldots (2)$$

Then by identical detailed procedure as that in Appendix II, the coefficients of Equation (2) are determined, and yield the following values:

$A = \theta_2$

$B = O$

$C = \frac{\theta_2m^2}{2}$

$D = \frac{\theta_2m^3}{6r_2}$

$E = \theta\left(\frac{m^3}{8r_2^2} + \frac{m^2}{24}\right)$ ..... etc.

Substituting the preceding values of the determined coefficients in Equation (2)

$$\theta = \theta_2\left[1 + \left(\frac{m^2}{2}\right)x^2 + \left(\frac{m^3}{6r_2}\right)x^3 + \left(\frac{m^3}{8r_2^2} + \frac{m^4}{24}\right)x^4\right.$$

$+ \ldots\ldots \Big] \ldots\ldots\ldots\

cumferential component of the fin temperature gradient. If the fin quadrant is sub-divided into a number of sectors bounded by radii from the tube center, then the error in assuming radial heat flow in any sector is reduced. Obviously, this error decreases with an increase in the number of sectors selected.

Consider a fin quadrant of length $(\frac{a}{2})$, width $(\frac{c}{2})$ and fin thickness $(b)$. Let the quadrant be sub-divided into $N$ sections such that $\frac{N}{2}$ sections are formed from equal segments of both its length and width. Then, depending on the value of $N$, each section may be approximately represented by a sector of radius $(r_2)$. In the forthcoming method the sectors are assumed as actually representing the fin plate, since, with the number of sectors chosen, the sector areas very nearly represent the actual section areas. Radial heat flow is assumed in all sectors. This assumption is only exact if the fin were slitted between sectors.

Fig. 7 illustrates a typical quadrant arbitrarily subdivided into eight sectors $(N = 8)$. The sector radii $(r_2)$ are computed as follows:

$$(r_2)_1 = \sqrt{\left(\frac{c}{2}\right)^2 + \left(\frac{m}{2}\right)^2} \quad \ldots\ldots (1)$$

$$\text{But } c = \frac{a}{T} \quad \ldots\ldots (2)$$

$$\text{And } m = \frac{a}{N} \quad \ldots\ldots (3)$$

Substituting values of $c$ and $m$ from Equations (2) and (3), respectively, in Equation (1) and simplifying, we find

$$(r_2)_1 = \frac{a}{2}\sqrt{\left(\frac{1}{T}\right)^2 + \left(\frac{1}{N}\right)^2} \quad \ldots\ldots (4)$$

Similarly,

$$(r_2)_2 = \frac{a}{2}\sqrt{\left(\frac{1}{T}\right)^2 + \left(\frac{3}{N}\right)^2} \quad \ldots\ldots (5)$$

$$(r_2)_3 = \frac{a}{2}\sqrt{\left(\frac{1}{T}\right)^2 + \left(\frac{5}{N}\right)^2} \quad \ldots\ldots (6)$$

and

$$(r_2)_4 = \frac{a}{2}\sqrt{\left(\frac{1}{T}\right)^2 + \left(\frac{7}{N}\right)^2} \quad \ldots\ldots (7)$$

In like manner, from inspection of Fig. 7, and using the relation:

$$n = \frac{c}{N} \quad \ldots\ldots (8)$$

We obtain:

$$(r_2)_5 = \frac{a}{2}\sqrt{1 + \left(\frac{7}{NT}\right)^2} \quad \ldots\ldots (9)$$

$$(r_2)_6 = \frac{a}{2}\sqrt{1 + \left(\frac{5}{NT}\right)^2} \quad \ldots\ldots (10)$$

$$(r_2)_7 = \frac{a}{2}\sqrt{1 + \left(\frac{3}{NT}\right)^2} \quad \ldots\ldots (11)$$

and

$$(r_2)_8 = \frac{a}{2}\sqrt{1 + \left(\frac{1}{NT}\right)^2} \quad \ldots\ldots (12)$$

The radius ratio of the first sector is then

$$R_1 = \frac{(r_2)_1}{r_1} \quad \ldots\ldots (13)$$

and values of $R$ for all other sectors are similarly obtained.

The fin plate area $(S)$ of each sector is computed as follows:

$$S_1 = [(r_2)_1^2 - r_1^2] \tan^{-1}\left(\frac{2m}{c}\right) \quad \ldots\ldots (14)$$

Where $\tan^{-1}\left(\frac{2m}{c}\right)$ is expressed in radians.

Substituting values of $c$, $m$, and $R$, from Equations (2), (3) and (13), respectively, in Equation (14) and simplifying

$$S_1 = r_1^2\ [R_1^2 - 1]\ \tan^{-1}\left(\frac{2T}{N}\right) \quad \ldots\ldots (15)$$

Similarly,

$$S_2 = r_1^2\ [R_2^2 - 1]\left[\tan^{-1}\left(\frac{4T}{N}\right) - \tan^{-1}\left(\frac{2T}{N}\right)\right] \quad \ldots\ldots (16)$$

$$S_3 = r_1^2\ [R_3^2 - 1]\left[\tan^{-1}\left(\frac{6T}{N}\right) - \tan^{-1}\left(\frac{4T}{N}\right)\right] \quad \ldots\ldots (17)$$

$$S_4 = r_1^2\ [R_4^2 - 1]\left[\tan^{-1}\left(\frac{8T}{N}\right) - \tan^{-1}\left(\frac{6T}{N}\right)\right] \quad \ldots\ldots (18)$$

By the same procedure, with Equation (8) replacing Equation (3) we obtain:

$$S_5 = r_1^2\ [R_5^2 - 1]\left[\tan^{-1}\left(\frac{8}{NT}\right) - \tan^{-1}\left(\frac{6}{NT}\right)\right] \quad \ldots\ldots (19)$$

$$S_6 = r_1^2\ [R_6^2 - 1]\left[\tan^{-1}\left(\frac{6}{NT}\right) - \tan^{-1}\left(\frac{4}{NT}\right)\right] \quad \ldots\ldots (20)$$

$$S_7 = r_1^2\ [R_7^2 - 1]\left[\tan^{-1}\left(\frac{4}{NT}\right) - \tan^{-1}\left(\frac{2}{NT}\right)\right] \quad \ldots\ldots (21)$$

and

$$S_8 = r_1^2\ [R_8^2 - 1]\ \tan^{-1}\left(\frac{2}{NT}\right) \quad \ldots\ldots (22)$$

Knowing $R$ for each sector, and with a given value of $P$, the fin effectiveness $E$ can be determined directly from Fig. 5.

The heat transferred through the first sector is

$$q_1 = hS_1E_1\theta_1 \quad \ldots\ldots (23)$$

so that the total heat transmission through all sectors is

$$q_8 = \Sigma q_{1\to 8} = h\theta_1\ [S_1E_1 + S_2E_2 + \ldots\ldots + S_8E_8]. \quad (24)$$

Let

$$q_8 = hS_8\bar{E}\theta_1 \quad \ldots\ldots (25)$$

and

$$S_8 = S_1 + S_2 + \ldots\ldots + S_8 \quad \ldots\ldots (26)$$

Then the mean effectiveness of the fin surface is represented by:

$$\bar{E} = \frac{S_1E_1 + S_2E_2 + \ldots\ldots + S_8E_8}{S_8} \quad \ldots\ldots (27)$$

For the square fin, since $a = c$

it is only necessary to compute data for the first $\frac{N}{2}$ sectors, since the square, when bisected diagonally, consists of two identically disposed fin surface areas.

## Application

The above methods, for approximation of fin effectiveness ($E$), are illustrated for both a rectangular and square plate fin. The design proportions of fin length and width to tube size represent values that should not be exceeded in good practice. These two examples are selected to demonstrate the maximum error in fin effectiveness ($E$) as encountered in extended surface design.

*Example* 1. Rectangular plate fin:

$a = 2\ 1/2$ in. or 0.208 ft
$c = 1\ 1/2$ in. or 0.125 ft
$r_1 = 5/16$ in. or 0.0260 ft
$k = 220$ Btu/(hr) (sq ft) (deg F/ft)
$h = 10$ Btu/(hr) (sq ft) (deg F)
$b = 0.010$ in. or 0.000833 ft

$$P = r_1\sqrt{\frac{2h}{bk}} = 0.026\sqrt{\frac{2 \times 10}{220 \times 0.000833}} = 0.272$$

$N = 8$ sectors/quadrant

$$T = \frac{a}{c} = 1.67$$

*(a) Estimation of fin effectiveness by equivalent circular area method:*

$$r_2 = \sqrt{\frac{ac}{\pi}} = \sqrt{\frac{0.208 \times 0.125}{\pi}} = 0.0911 \text{ ft}$$

$$R = \frac{r_2}{r_1} = \frac{0.0911}{0.0260} = 3.58$$

and

$P = 0.272$

Hence

$\bar{E} = 0.772$ (from Fig. 5)

*(b) Estimation of fin effectiveness by segmentation:* Referring to equations of Method No. 2 the data for sector 3 are computed in detail

$$(r_2)_3 = \frac{a}{2}\sqrt{\left(\frac{1}{T}\right)^2 + \left(\frac{5}{N}\right)^2}$$

or

$$(r_2)_3 = \frac{0.208}{2}\sqrt{\left(\frac{1}{1.67}\right)^2 + \left(\frac{5}{8}\right)^2} = 0.0901 \text{ ft}$$

$r_1 = 0.0260$ ft

$$R_3 = \frac{(r_2)_3}{r_1} = \frac{0.0901}{0.0260} = 3.47$$

$$S_3 = r_1^2\ [R_3^2 - 1]\left[\tan^{-1}\left(\frac{6T}{N}\right) - \tan^{-1}\left(\frac{4T}{N}\right)\right]$$

or

$$S_3 = 0.0260^2\ [3.47^2 - 1]$$

$$\left[\tan^{-1}\frac{6 \times 1.67}{8} - \tan^{-1}\frac{4 \times 1.67}{8}\right] = 0.00150 \text{ sq ft}$$

Data for other sectors are computed in an identical manner.

The following table lists essential data to determine the mean fin effectiveness, $\bar{E}$.

| Sector No. | R | P | S | E | SE | $\bar{E}$ |
|---|---|---|---|---|---|---|
| 1..... | 2.45 | 0.272 | 0.00135 | 0.924 | 0.00125 | — |
| 2..... | 2.84 | 0.272 | 0.00143 | 0.879 | 0.00126 | — |
| 3..... | 3.47 | 0.272 | 0.00150 | 0.788 | 0.00118 | — |
| 4..... | 4.23 | 0.272 | 0.00154 | 0.675 | 0.00104 | — |
| 5..... | 4.52 | 0.272 | 0.00155 | 0.635 | 0.00098 | — |
| 6..... | 4.27 | 0.272 | 0.00154 | 0.670 | 0.00103 | — |
| 7..... | 4.10 | 0.272 | 0.00153 | 0.693 | 0.00106 | — |
| 8..... | 4.02 | 0.272 | 0.00153 | 0.705 | 0.00108 | — |
| Σ..... | .... | ..... | 0.01197 | ..... | 0.00888 | 0.742 |

The mean fin effectiveness, for this fin, as computed by the equivalent circular area method, is high by approximately

$$\frac{(0.772 - 0.742)\ 100}{0.742} = 4 \text{ per cent}$$

This amount of error is probably within limits of accuracy involved in the assumption of a constant fluid convection coefficient ($h$) over the entire fin surface, as used in deriving the expression for fin effectiveness. For practical purposes, the use of the equivalent circular area method gives a sufficiently accurate value of fin effectiveness, although this results in a somewhat higher value than indicated by the more exact method.

*Example 2*—Square plate fin:

$a = 2$ in. or 0.167 ft
$c = 2$ in. or 0.167 ft
$r_1 = 5/16$ in. or 0.0260 ft
$k = 220$ Btu/(hr) (sq ft) (deg F/ft)
$h = 10$ Btu/(hr) (sq ft) (deg F)
$b = 0.010$ in. or 0.000833 ft

$$P = r_1\sqrt{\frac{2h}{bk}} = 0.272$$

$N = 8$ sectors/quadrant

$$T = \frac{a}{c} = 1$$

*(a) Estimation of fin effectiveness by equivalent circular area method:*

$$r_2 = \sqrt{\frac{ac}{\pi}} = \sqrt{\frac{0.167 \times 0.167}{\pi}} = 0.0940 \text{ ft}$$

$$R = \frac{r_2}{r_1} = \frac{0.094}{0.026} = 3.61$$

and

$P = 0.272$

Hence

$E = 0.768$ (from Fig. 5)

*(b) Estimation of fin effectiveness by segmentation:* The following table lists necessary factors for determining the mean fin effectiveness and are computed as in *Example 1.*

| Sector No. | R | P | S | E | SE | $\bar{E}$ |
|---|---|---|---|---|---|---|
| 1..... | 3.23 | 0.272 | 0.00156 | 0.824 | 0.00128 | — |
| 2..... | 3.43 | 0.272 | 0.00160 | 0.795 | 0.00127 | — |
| 3..... | 3.78 | 0.272 | 0.00162 | 0.743 | 0.00120 | — |
| 4..... | 4.25 | 0.272 | 0.00163 | 0.672 | 0.00110 | — |
| Σ..... | .... | ..... | 0.00641 | ..... | 0.00485 | 0.757 |

The mean fin effectiveness, as computed by the equivalent circular area method, is high by approximately

$$\frac{(0.768 - 0.757)\ 100}{0.757} = 1.5 \text{ per cent}$$

The method using the equivalent circular area therefore indicates a fin effectiveness of sufficient accuracy to use in practice, when applied to square fin plates.

*Note:* The foregoing method of determining the fin effectiveness by segmentation, based on the assumption of a square or rectangular fin area served by each tube, applies to both *in line* or *staggered* tube arrangements where *individual* fins are assembled on *each* tube. Further, it is applicable to *in line* tube nests, as shown in Fig. 4, where the fin plate is *continuous* over the entire tube nest.

In designs employing a *staggered* tube arrangement with a *continuous* fin plate, the fin area served by a tube is hexagonal. For this type surface arrangement the method of rectangular or square fin segmentation, developed in this appendix, may be applied in a similar manner to the hexagonal area.

## Appendix V

## Examples Illustrating the Relative Magnitude of Individual Thermal Resistances of a Typical Extended Surface Air Cooler for Various Applications

$S = S_t + S_p$ = Total outside surface, sq ft/lineal ft tube
$S_i$ = Inside tube surface, sq ft/lineal ft tube
$S_p$ = Outside exposed tube surface, sq ft/lineal ft tube
$S_t$ = Lateral fin surface, sq ft/lineal ft tube
$a$ = Fin plate length/tube, ft
$b$ = Fin plate thickness, ft
$c$ = Fin plate width/tube, ft
$E$ = Fin effectiveness, dimensionless
$F_s$ = Ratio of sensible to total heat removed from air, dimensionless
$h_a$ = Air film coefficient, Btu/(hr) (sq ft of $S$) (deg F)
$h_i$ = Water film coefficient, Btu/(hr) (sq ft of $S_i$) (deg F)
$k$ = Fin metal thermal conductivity, Btu/(hr) (sq ft) (deg F/ft)

$P = r_1\sqrt{\frac{2h}{kb}}$, dimensionless

$r_1$ = Inside tube radius, ft
$r_1$ = Outside tube radius, ft

$r_2 = \sqrt{\frac{ac}{\pi}}$ = Radius of equivalent circular fin area, ft

$R = r_2/r_1$, dimensionless
$R_{os}$ = Overall thermal resistance to sensible heat flow, Btu/(hr) (sq ft of $S$) (deg F overall)

$R_a = \frac{1}{h_a}$ = Air film thermal resistance, Btu/(hr) (sq ft of $S$) (deg F)

$R_m = \frac{1}{h_a}\left(\frac{1-E}{E+\frac{S_p}{S_t}}\right)$ = Metal thermal resistance, Btu/(hr) (sq ft of $S$) (deg F)

$R_i = \frac{S}{S_i h_i}$ = Water film thermal resistance, Btu/(hr) (sq ft of $S$) (deg F)

*Surface Used for Illustration:* A typical, extended surface air cooler, used commercially, has been selected for illustrative purposes. The fin surface consists of rectangular, flat plates bonded to round tubes. In the following examples air, flowing over the fin plates, is cooled by water passing through the tubes. The surface is nested with tubes *in line* in adjacent rows.

The surface physical data are as follows:

$a$ = 1-11/16 in. or 0.141 ft
$b$ = 0.010 in. or 0.000833 ft
$c$ = 1-1/2 in. or 0.125 ft
$r_1$ = ¼ in. or 0.0209 ft
$r^1$ = 0.222 in. or 0.0185 ft

$r_2 = \sqrt{\frac{ac}{\pi}}$ = 0.0749 ft

$R = r_2/r_1 = 3.60$
$S_i$ = 0.116 sq ft/lineal ft tube
$S_t$ = 2.33 sq ft/lineal ft tube
$S_p$ = 0.12 sq ft/lineal ft tube
$S$ = 2.45 sq ft/lineal ft tube
*Fin Spacing* = 6/in.
*Tube Material* = Copper
*Fin Material* = Copper

*Thermal Resistance Formula:* In the following formula for the overall sensible heat thermal resistance ($R_{os}$), the thermal resistance through the prime tube wall is neglected, and a perfect thermal bond assumed between the fin base and outside tube wall. The individual thermal resistances, referred to the total outside surface area ($S$), form the overall sensible heat thermal resistance ($R_{os}$) as follows:

$$R_{os} = R_a + \frac{R_m + R_i}{F_s} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

*Scope of Examples: Examples 1* and *2* compare the typical heat transfer performance of free versus forced air convection applications, respectively. No air latent heat removal is considered in either of these examples.

*Example 3* indicates the heat transfer performance where some moisture is removed from the chilled air. This latter example represents a normal application for surface dehumidifiers.

*Value of tube and air side transfer coefficients:* The tube side (water) heat transfer coefficient ($h_i$) is selected for approximately 3 ft per second water velocity at normal chilled water temperatures. The two values of the air film transfer coefficient ($h_a$) illustrate the difference between free and forced air convection applications. Table 1 lists the values of both type coefficients used in these comparisons.

**Table 1—Value of Both Types of Transfer Coefficients Used**

| Example No. | 1 | 2 | 3 |
|---|---|---|---|
| $h_a$ | 4 | 9 | 9 |
| $h_i$ | 600 | 600 | 600 |

*Evaluation of metal thermal resistance* ($R_m$): The fin effectiveness ($E$) is evaluated by using the equivalent circular area method. (See Appendix IV). The steps in determining both $E$ and $R_m$ are summarized in Table 2.

**Table 2**

| Example No. | 1 | 2 and 3 |
|---|---|---|
| $r_2$ | 0.0749 ft | 0.0749 ft |
| $r_1$ | 0.0208 ft | 0.0208 ft |
| $h_a$ | 4 | 9 |
| $k$ | 220 | 220 |
| $b$ | 0.000833 ft | 0.000833 ft |
| $P = r_1\sqrt{\frac{2h_a}{bk}}$ | 0.138 | 0.207 |
| $R = r_2/r_1$ | 3.60 | 3.60 |
| $E$ from Fig. 5 | 0.925 | 0.849 |
| $S_p$ | 0.12 | 0.12 |
| $S_t$ | 2.33 | 2.33 |
| $R_m$ | 0.0192 | 0.0186 |

Note from Table 2 the small variation of $R_m$ over a wide range of the air film coefficient ($h_a$). For this specific fin, $R_m$ changes approximately 3 per cent for a 225 per cent change of $h_a$, while $E$ varies about 9 per cent.

*Relative magnitudes of individual thermal resistances for various applications:* Table 3 is compiled from use of Equation (1) and data in Tables 1 and 2.

**Table 3**

| Example No | 1 | 2 | 3 |
|---|---|---|---|
| Application | Free Conv. (Dry) | Forced Conv. (Dry) | Forced Conv. (Dehumidifying) |
| $F_s$ | 1.0 | 1.0 | 0.5 |
| $R_a$ | 0.250 | 0.111 | 0.111 |
| $R_m/F_s$ | 0.0192 | 0.0186 | 0.0266 |
| $R_i/F_s$ | 0.0352 | 0.0352 | 0.0508 |
| $R_{os}$ | 0.3044 | 0.1648 | 0.1879 |
| 100 ($R_a/R_{os}$) = per cent air film resistance | 82.1 | 67.3 | 59.0 |
| 100 ($R_m/R_{os}\times F_s$) = per cent metal resistance | 6.3 | 11.3 | 14.2 |
| 100 ($R_i/R_{os}\times F_s$) = per cent water film resistance | 11.6 | 21.4 | 26.8 |

The last three lines in Table 3 list the values of the individual thermal resistances, expressed as a percentage of the overall sensible heat thermal resistance. The metal or fin plate represents from approximately 6 per cent to 14 per cent of the total resistance to heat flow, for this specific coil design.

Thus a knowledge of the metal resistance is of importance in determining the thermal performance of extended surfaces. The metal resistance may or may not be an appreciable part of the total thermal resistance depending upon combinations of the following design variables for the specific transfer surface:

1. $r_1$
2. $r_2$
3. $h_a$
4. $b$
5. $k$
6. $h_i$
7. $S/S_i$
8. $F_s$

## Appendix VI

### Comparison of Heater Performance

It is frequently required to compare the performance of one heater with another of different design or proportions. These heaters are generally tested at different air velocities but generally the temperature rise and the resistance to air flow will not correspond. The values for $U$ or the

overall rate of heat transfer may or may not be the same. The commercial value of a heater per square foot of surface depends upon the value of $U$ under some fixed condition at which it is supposed to operate. These fixed conditions are ratio of initial to final temperature difference and pressure drop when handling a fixed quantity of air. The usual method of approach to this problem has been to draw graphs of performance and interpolate for corresponding conditions for two different heaters. This has been well outlined by Stacey and Ashley[6] but it is also possible to do this without graphs where $h$, the coefficient of heat transfer, and the pressure drop, $\triangle p$, for a given arrangement of heater and air quantity, is known. In comparing the economy of two different designs of plate fins, for example, it is necessary to have a covenient method for making this comparison. Such a method has been worked out and has been in use, in effect, for some time. Since the use of this method is an essential part in the design study of heaters, its presentation here is perhaps in order. The following symbols are employed:

$S$ = square foot of heater surface
$A$ = face area of heater—square feet
$L$ = relative depth of heater
$G$ = pounds of air per hour passing through the heater
$V$ = air velocity through face area of heater
$C_p$ = specific heat of the air or change in total heat content per degree change in temperature
$\rho$ = air density—pounds per cubic foot
$\triangle p$ = drop in air pressure passing through the heater at the velocity $V$—inches water gauge
$t_s$ = temperature of heating or cooling fluid
$t_1$ = temperature of entering air
$t_2$ = temperature of leaving air
$\theta_1 = t_s - t_1$
$\theta_2 = t_s - t_2$
$R = \left( \frac{\theta_1}{\theta_2} \right)$
$U$ = overall coefficient of heat transfer Btu per hour per square foot per degree difference

Perhaps the simplest method of approach to the above problem is as follows: Assume a heat transfer surface of flexible arrangement as to depth ($L$) and area ($A$) but constant surface ($S$) so that $AL$ is constant for all arrangements. We may now show that by change in $A$, $L$ and $G$ we may simulate any performance in respect to $R$ and $\triangle p$.

There are three primary equations which relate the independent variables $\triangle p$, $R$ and $V$. These are:

$$\log R = \frac{US}{C_p G} \quad \ldots\ldots (1)$$

$$\triangle p = C_1 V^n L \quad \ldots\ldots (2)$$

*where*

$C_1$ and $n$ = constants for any given type of heater surface, $n$ = a value approaching 2 but always slightly less than 2.

$$U = C_2 V^m \quad \ldots\ldots (3)$$

*where*

$C_2$ and $m$ = empirical values determined by test for a given type of heater, and may both be taken as constant through a velocity range not exceeding a ratio of 2 to 1.

Also, we have the following dimensional relationships based on the assumption that the surface $S$ remains constant but that the face area $A$ and the depth of the heater $L$ are varied to simulate any desired performance in respect to $R$ and $\triangle p$.

$AL = C_3$, a constant since $S$ is constant for above assumption $\ldots\ldots (4)$

$$V = \frac{G}{60\rho A} \text{ or } G = 60AV\rho \quad \ldots\ldots (5)$$

Since we are to compare two arrangements which we may designate as (1) and (2), we will employ the ratios of the various values rather than the absolute values. For convenience, we may express the ratios

$\frac{\log R_2}{\log R_1}$, $\frac{\triangle p_2}{\triangle p_1}$, $\frac{A_2}{A_1}$, etc., simply as $\log R_2$, $\triangle p_2$, $A_2$, etc. Then we have from Equations (1), (2) and (3)

$$\log R_2 = \frac{U_2}{G_2} \quad \ldots\ldots (6)$$

$$\triangle p_2 = V_2^n L_2 \text{ and} \quad \ldots\ldots (7)$$

$$U_2 = V_2^m \quad \ldots\ldots (8)$$

From Equations (4) and (5) we have

$$A_2 L_2 = 1 \quad \ldots\ldots (9)$$

and

$$G_2 = A_2 V_2 \quad \ldots\ldots (10)$$

Combining Equation (6) with Equation (10), we have

$$\log R_2 = \frac{V_2^m L_2}{V_2} = \frac{L_2}{V_2^{(1-m)}} \quad \ldots\ldots (11)$$

or

$$L_2 = V_2^{(1-m)} \log R_2 \quad \ldots\ldots (12)$$

From Equation (7), we have

$$L_2 = \frac{\triangle p_2}{V_2^n} \quad \ldots\ldots (13)$$

Equating Equation (12) and Equation (13), we have

$$V_2^{(1-m)} \log R_2 = \frac{\triangle p_2}{V_2^n} \quad \ldots\ldots (14)$$

$$V_2^{(1+n-m)} = \frac{\triangle p_2}{\log R_2} \quad \ldots\ldots (15)$$

Combining Equation (15) with Equation (8), we have

$$U_2 = \left( \frac{\triangle p_2}{\log R_2} \right)^{m/(1+n-m)} \quad \ldots\ldots (16)$$

That is

$$\frac{U_2}{U_1} = \left[ \frac{\triangle p_2 / \triangle p_1}{\log R_2 / \log R_1} \right]^{m/(1+n-m)}$$

Also we find

$$\frac{(S/A)_2}{(S/A)_1} = \left( \frac{U_1}{U_2} \right) \left( \frac{V_2}{V_1} \right) \quad \ldots\ldots (17)$$

Equation (16) now gives us the comparative value of $U_2$ to $U_1$ for the desired values of $\triangle p_2$ and $R_2$. We can now compare the heater B directly with the heater A since they both have the same pressure drop and the same ratio of initial to final temperature difference. That is, they are compared under conditions of equal performance. If $U_a$ is the coefficient of heat transfer of heater A and $U_{b2}$ is the heat transfer coefficient for heater B under conditions of equal performance as in Equation (16), then the ratio $\frac{U_{b2}}{U_a}$ is the relative value of the two surfaces per square foot.

Generally, the exponents $m$ and $n$ can each be determined from two test points on heater B if these test points are determined with sufficient accuracy or lie on a test curve. The two test points chosen should, in each case, bracket the temperatures and pressures which are to be compared. If two test points of pressure $\triangle p_1$ and $\triangle p_3$ are determined for corresponding velocities $V_1$ and $V_3$, then the exponent $n = \log \left( \frac{\triangle p_3}{\triangle p_1} \right) \Big/ \log \left( \frac{V_3}{V_1} \right)$ and the exponent for heat transfer $m = \log \left( \frac{U_3}{U_1} \right) \Big/ \log \left( \frac{V_3}{V_1} \right)$.

Thus while $m$ and $n$ may vary at different points of the curve, yet the values of $m$ and $n$ that apply to the correction in Equation (16) are obtained with sufficient accuracy so that the desired value $U_2$ or $\frac{U_2}{U_1}$ is as accurate as any test value would be for the heater in which the dimensions were changed. It is understood, of course, that $\triangle p_3$ and $U_3$ are determined directly from tests on heater B while $\triangle p_2$ and $U_2$ are hypothetical calculated values which by rearrangement of heater B would give the same results as that obtained by Heater A. An example will make this clear.

*Example:* Assume we have a heater which we will designate as design A for which we have tabulated performance values against which we wish to compare the correspond-

[6] Loc. cit. See Note 2.

ing performance of a different design, B; and that we have at least two accurate test points of performance for the latter. What is the relative utility value of a square foot of surface of B as compared with a square foot of surface of A? Assume we select one point of performance of heater A which is bracketed by the two test points for heater B and that we have the following performance data:

For A—

$V_A = 500$ ft/min = velocity of standard air over face area.

$(\Delta p)_A = 0.4$ in.

$\theta_{1A} = t_S - t_1 = 100$ = difference between the temperature of the heating fluid and the entering air

$\theta_{2A} = t_S - t_2 = 30$ = difference in temperature between the heating fluid and the leaving air

$$R_A = \left(\frac{\theta_{1A}}{\theta_{2A}}\right) = 3.33$$

$C_p = 0.24$

$U_A = 10$

$(S/A)_A = 65.0$

*First Test Point (for B)*

$V_1 = 540$ ft/min

$\Delta p_1 = 0.5$ in. (water gauge)

$\theta_1 = 100$ degrees

$\theta_2 = 40$ degrees

$$R_1 = \left(\frac{\theta_1}{\theta_2}\right) = 2.5$$

$C_p = 0.24$

$(S/A)_B = 56.4$ sq ft/sq ft

$$U_1 = \frac{60 C_p \rho V_1 \log R_1}{(S/A)_B} = \frac{1.08 \times 540 \times 0.915}{56.4} = 9.47$$

*Second Test Point (for B)*

$V_2 = 360$

$\Delta p_2 = 0.24$ in.

$\theta_1 = 90$ degrees

$\theta_2 = 30$ degrees

$$R_2 = \left(\frac{\theta_1}{\theta_2}\right) = 3.0$$

$C_p = 0.24$

$(S/A)_B = 56.4$

$$U_2 = \frac{1.08 \times 360 \times 1.099}{56.4} = 7.58$$

*From* B—*First and Second Tests*

$$\left(\frac{V_2}{V_1}\right)^m = \frac{U_2}{U_1} = \left(\frac{360}{540}\right)^m = \left(\frac{7.58}{9.47}\right) = 0.80$$

$m = .55$

$$\left(\frac{V_2}{V_1}\right)^n = \frac{\Delta p_2}{\Delta p_1} = \left(\frac{360}{540}\right)^n = \left(\frac{0.24}{0.50}\right)$$

$n = 1.805$

Having thus established $m = 0.55$ and $n = 1.805$ from the two tests for design B, and taking from the first test point $R_B = \left(\frac{\theta_1}{\theta_2}\right) = 2.5$, $\Delta p_B = 0.5$ in., $U_B = 9.47$, $V_B = 540$, we may calculate $U_{BC}$ and $V_{BC}$ for a new arrangement $(S/A)_{BC}$ which will give $R_{BC} = R_A = 3.33$ and $(\Delta p)_{BC} = (\Delta p)_A = 0.4$ in.

The equation giving $U_{BC}$ (see Equation 16) is

$$\frac{U_{BC}}{U_{B1}} = \left[\frac{(\Delta p)_{BC}/(\Delta p)_{B1}}{\log R_{BC}/\log R_{B1}}\right]^{m/(1+n-m)} = \left(\frac{0.80}{1.33}\right)^{0.244} = 0.883$$

$U_{BC} = 9.47 \times 0.883 = 8.36$

We are now able to compare the value of the B surface with that of the A surface. This ratio of value under conditions of equal performance is

$$100\left(\frac{8.36}{10}\right) = 83.6 \text{ per cent} = \frac{\text{Value of sq ft B surface}}{\text{Value of sq ft A surface}}$$

$$V_{BC} = 540 \times \left(\frac{8.36}{9.47}\right)^{(1/0.55)} = 430 \text{ ft/min}$$

Corresponding to Equation (17), the formula giving $(S/A)_{BC}$ is

$$(S/A)_{BC} = (S/A)_A \times \left(\frac{V_{BC}}{V_A}\right) \times \left(\frac{U_A}{U_{BC}}\right)$$

$$(S/A)_{BC} = 65.0 \times \left(\frac{430}{500}\right) \times \left(\frac{10}{8.36}\right) = 66.9 \text{ sq ft/ sq ft}$$

## Discussion

H. B. Nottage, East Hartford, Conn.: Some time ago, when Dr. Carrier was engaged in completing the first formulations of the problem discussed in this paper, it was my fortunate privilege to be given the opportunity of reviewing the analytical details involved. The series solution to the basic differential equation most definitely seemed to offer practical possibilities of effective application.

My particular interest dealt with a general approach to the case of a circumferential fin with tapered sides, on a base cylinder of fairly large diameter. In further regard, I was concerned with the design problem for a family of rather closely-spaced fins, rather than an individual isolated fin. It is to be recognized that there is no exact analytical solution available for the basic differential equation describing this case. A series solution offers one type of approximation method; but the use of a series implies an obligation to also bring forth a concise coverage of the limitations on range and accuracy inherent in all such approximations. No doubt, the authors' solution is adequate for most engineering purposes, and what are its limitations?

Following my initial discussion with Dr. Carrier, I was fortunate enough to procure the services of a physicist and a mathematician in seeking a precise evaluation of the merits of the series solution. The net result of these efforts was a judgment that the series method was adequate only for fins of fairly high effectiveness and slight taper. For the general case, there was recommended to me, instead, a solution of the basic differential equation based upon Picard's method of successive approximations. When applied to the problem of my particular interest, this latter method was demonstrated to have a more rapid convergence and improved accuracy over the series solution; although this is no serious point against the series method when properly delimited.

Further comments on the assumptions invoked by the authors are in order.

It was stated that the solution obtained required the fin base temperature to be known, and it was further implied that this temperature should be the same at all points. I do not necessarily feel that such a limitation is essential. It is possible to extend the solution of the problem by eliminating the base temperature and its variations from point to point, in terms of the independent variables which establish this temperature. This extension involves going into the fluid conditions on the prime-surface side of the exchanger, and includes a consideration of the heat conduction through the base metal. The heat transfer from the exposed surface of the base metal is also properly included through this extension of the general problem; and this term is not negligible in many instances.

Another serious limitation on the extensive application of this analysis.

which is common to most of those which are set up to deal with fin problems, lies in the assumption that the unit surface conductance for heat transfer is constant at some mean effective value over the entire surface. This is done for want of adequate data on point-to-point conditions for the heat transfer and flow processes. The effect is usually blanketed out in the reduction of over-all experimental data, followed by the re-application of such data to design problems on an over-all basis. However, it has been my experience that this unit surface conductance *cannot* be treated as a mean constant quantity over a wide range of geometrical and heat transfer conditions. There is a very real point-to-point variation here, which must be recognized and accounted for, to permit design to close limits over all extremes of operating conditions. I can not emphasize too strongly the present need for patient and thorough experimental data covering problems of this nature.

The idealization that all of the heat flow through the fins is radially outward requires the isotherms in the metal to be concentric circles. This would not be expected always; although apparently the analysis is not too sensitive to departures therefrom. The mathematics of combined radial and circumferential heat flow is difficult for all but the simpler cases.

Then, there arises the question of performance comparisons. The authors have dealt with over-all comparisons only. These regard the heat exchanger as a *box*, through which streams flow, and for which certain hypotheses are made regarding the internal heat exchange process. Then, convenient empirical exponents are applied, based upon data which are usually rather limited and not necessarily too accurate. These methods are workable for the usual interpolation comparisons and quick estimates; but for design problems where extrapolation is required, a breakdown into the component effects influencing the over-all performance is desirable. It is my impression that often empirical exponents are trusted too far under inappropriate circumstances.

When we come to comparisons dealing with the merits of the component process of an over-all heat exchange, a first interest is in an evaluation of the performance of the fluid-swept surface itself, in its form, finish, and extent. For instance, the crimped fin mentioned by the authors has an irregular or rough surface. How is a true judgment of this construction to be obtained upon an internal rather than an over-all basis? A method of doing this has recently been proposed by Dr. A. P. Colburn[7]. In effect, his suggested comparison interrelates the unit surface conductance for heat transfer and the corresponding power loss in the fluid flow stream per unit of surface area swept. This, I have found very useful, myself.

By way of conclusion, I would like to compliment the authors for their contributions to an important problem. The fin effectiveness correlation presented in this paper should aid greatly the understanding of such extended-surface heat exchanger units as are employed in heating, ventilating, and air conditioning equipment.

G. L. Tuve, Cleveland, Ohio: This sounds a bit complicated, but actually it is a simplification of the problem compared with what we have had before, and I think the authors are to be complimented. I hope it will work out so that the three types of resistance that we recognize in a fin coil can all be taken care of easily. We have an air side resistance, a fin effectiveness, and an inside surface coefficient resistance. The air side resistance problem has been pretty well aired and after the war, when it can be discussed, I think that will be cleared up.

Nobody knew just what to say about fin effectiveness. The authors are to be complimented on simplifying this phase of the problem. The inside surface coefficient is complicated for one case; namely, that of evaporating refrigerants. The Society is now initiating some research along that line that I hope may help to clear it up.

We are very much in need of a standard test code for blast coils, preferably one that is simpler than the code proposed by the *Blast Coil Manufacturer's Institute*. General agreement upon a simplified method for computing blast coil performance would be highly desirable, and this paper is an important contribution toward that goal.

William Goodman, LaCrosse, Wis.: To many of us the principal value of *fin effectiveness* is that it enables us to predict the performance of various coils in advance. To do so we must use the air film coefficient for prime surface and the fin effectiveness together. Then with the inside surface film coefficient known, the over-all film coefficient can be predicted. Similarly, to determine the true air film coefficient from a test on a finned coil, accurate values of the *fin effectiveness* are also required.

However, in order either to predict the over-all coefficient of a finned coil or to determine the actual value of the film coefficient from a test on a finned coil, we must be certain that the external film coefficient—rather, the air film coefficient, is not affected by the presence of the fins. In some tests reported in a European paper a few years ago it was found that fin spacing did have a marked effect on the film coefficient. As long as the fins were widely spaced along the tube the film coefficient was not affected by the presence of the fins, and therefore, the effectiveness could be used to predict quite accurately the over-all film coefficient; but when the fins were crowded together and the spacing between the fins was reduced to less than approximately one-third of the tube diameter, then the results began to disagree. The over-all coefficient determined by experiments was usually lower than the result obtained by computation.

In all our work, at least in commercial coils, the fins are much closer than one-third of the tube diameter. They are frequently considerably less than that, so that in applying the fin effectiveness we have to be careful.

I do not mean in any way to detract from the authors' work, because the accurate determination of fin effectiveness is an important step, and one which they seem to have accomplished, but I do think that further work is necessary in applying these fin effectiveness values to closely spaced fins.

Dr. Carrier: With reference to Mr. Nottage's comments, I was very much interested in the fact that he had referred this to mathematicians and physicists because it is really quite a difficult mathematical and physical problem. I note that he says they questioned the accuracy of the series method of solution. Qualitatively the mathematician would be quite correct as, for example, in the series of values shown by the curves. If we were to use values beyond that shown in the curves or even the more extreme values given in the curves, the error might be appreciable if an insufficient number of terms were employed without a correction for the remaining terms. Fortunately, however, the values in which we are interested from a practical standpoint never lie in this extreme region. Furthermore, we did take particular precautions to investigate the probable residual value of the remaining terms. In every instance, we calculated eight terms and in a few cases to determine the trend of the residual we calculated nine or ten terms. A rather ingenious method was also worked out which seemed to predict quite accurately the summation of the remaining terms in the infinite series. This method appeared to be quite accurate so long as the series remained reasonably convergent. This method depended upon a study of the decrements of the unknown residual with successive terms up to ten. The law of decrease was found to be a straight line logarithmic function within the range that we could test it. Therefore, it became possible to integrate the residual mathematically on the assumption that this logarithmic decrement continued. This, of course, assumed an extrapolation which could not be proven beyond the ten terms examined. However, in the calculation of all practical physical arrangements the value of this decrement was so small as to not be of serious consequence if it were omitted and the error in its calculation could in no way be large. The curves which are published in this paper contain the correction for the residual and there

[7]Heat Transfer by Natural and Forced Convection, by A. P. Colburn. (*Purdue University Engineering Bulletin*, Vol. XXVI, No. 1, January, 1942.)

is no possibility by any other method of calculation of finding results which will differ from those shown by more than an insignificant percentage. In other words, within the limits shown the results are correct by any method of integration.

There was another point of considerable interest that was made by Mr. Nottage. This was the effect that the varying values of $h$ (surface coefficient of heat transfer) might have on the apparent metal resistance since the value of $h$ appears in equation of the metal resistance. It is true, of course, that a change in the value of $h$ at different parts of the surface would distort the lines of heat flow as would also the change in temperature of the air passing over the surface. The distorted lines of heat flow would be longer than the straight lines and not radial as assumed. The answer to this is that in calculating the resistance from the effectiveness curves, the value of $h$ appears both in the numerator and the denominator of a fraction so that in calculating the resistance of the metal on the basis of the resistance formula, variations in $h$ largely iron out or cancel, although not entirely. It is for this reason that approximate test values of $h$ or even approximately assumed values of $h$ do not greatly change the calculated value of resistance of the fins although they do change considerably the value of a calculated effective temperature difference.

It would perhaps have been better to have drawn a plot in terms of resistance rather than in terms of effective temperature difference since the former would show but little change with the value of $h$. The average values of $h$ are, of course, determined as Mr. Goodman indicates from actual tests in which other variables may be evaluated and then eliminated. This procedure gives us practical average values of $h$ which do not vary widely for small changes in design. The greatest change would probably be caused by the change in the spacing of the fins and for any large change in spacing the value of $h$ would have to be determined by test unless the flow in all cases was in the nonturbulent region. In the latter case, $h$ would vary approximately inversely as the spacing. In the turbulent region, the value of $h$ should follow closely the square root of the coefficient of friction which can be calculated with reasonable accuracy from known data.

As for the method of comparison of the commercial value of two different types of heat transfer surface, there are as indicated in the paper two methods which may be employed. The older method, which is also quite correct, is by interpolation of plots in a family of curves for both types of heaters. One of these plots is superimposed on the other and the point of common performance is found on each. This method is correct but the method which I have proposed has a considerable advantage in comparing a single point of performance for the two units and is equally accurate if we assume no observational errors. It is well known, however, that different heaters will compare somewhat differently in different ranges of performance and if the entire range of performance is to be examined, then the method of calculation has but little if any advantage over the graphical method.

It should be observed that the over-all value of heat transfer $U$ is the only value that has any significance in comparing performance of two different types of units. In analysis for design, the heat transfer coefficients or resistance must be separated as component parts of the over-all value $U$ but it should be noted that a heater may have a very low metal resistance or a very high value of $h$ and still be a poor heater because of the remaining resistance being excessive. It is the comparison of the final designs in relation to their practical performance that alone has commercial importance.

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## ASHVE Research Laboratory

1. Case School of Applied Science
2. Western Reserve University Medical School and Hospitals
3. Severance Hall

**Aerial View of ASHVE Research Laboratory**

Work at the ASHVE Research Laboratory will go forward under a $50,000 budget for research and cooperative activities for the fiscal year under Director Cyril Tasker.

On April 29, a meeting of the Research Executive Committee was held with Chairman G. L. Tuve presiding.

An inspection of the Laboratory, located at 10700 Euclid Ave., Cleveland, was made by members of the Council of the Society on April 30.

Fourteen members of the Council were present and made a tour of the building, and this was followed by a reception for the visiting members tendered by the Officers and members of the Northern Ohio Chapter of the Society.

# SUMMARY OF LOCAL CHAPTER MEETINGS

| CHAPTER | MEETING DATE | SUBJECT | SPEAKERS | OTHER FEATURES | ATTENDANCE | ATTENDANCE RATIO* |
|---|---|---|---|---|---|---|
| Cincinnati | Mar. 14 | Heating, Ventilating, and Air Conditioning of Pentagon Building, Washington, D. C. | Charles S. Leopold, Consulting Engr., Philadelphia, Pa. | Committee reports submitted. G. V. Sutfin presented Chapter with Scrap Book for Library containing interesting items, papers, photos, etc., from 50th Annual Meeting of Society | 42 | 0.90 |
| Connecticut | Mar. 17 | The Flow of Air and Gas in Various Systems | S. D. Downs, Kalamazoo, Mich., Pres. of ASHVE | Report of Nominating Committee was read. | 52 | 0.54 |
| Delta | Mar. 21 | Air Conditioning and Lighting | Thomas H. Bean, Jr., New Orleans Public Service, Inc. | G. C. Kerr read a French Canadian poem "How Baptiste Came Home" | 30 | 0.50 |
| | | Joint meeting with New Orleans Chapter of IES as guests of Delta | | Pres. L. V. Cressy reported on A. V. Hutchinson's visit | | |
| Illinois | Mar. 13 | Housing Research—An Aid to Heating and Ventilating Engineers | C. F. Boester, Housing Research Exec., Purdue Research Foundation and Pres. of St. Louis Chapter | P. J. Marschall reported for Membership Committee<br>S. R. Lewis summarized work of Postwar Program Committee | 76 | 0.52 |
| Indiana | Mar. 31 | Elimination of Air from Heating Units | Ferdinand Jehle, Dir. of Research, Hoffman Specialty Co., Indianapolis, Ind. | C. F. Boester, Pres., St. Louis Chapter, spoke on Chapter Delegates Committee | 52 | 0.52 |
| | Feb. 25 | Smoke Abatement | Prof. R. R. Tucker, Washington University, St. Louis | Pres. S. E. Fenstermaker asked Carl Boester to introduce Professor Tucker | 70 | 0.77 |
| Iowa | Mar. 15 | Water Cooling Equipment | L. T. Mart, The Marley Co., Kansas City, Mo. | Mr. Mart illustrated his talk with many interesting slides | 24 | 0.96 |
| Kansas City | Mar. 27 | Air Recovery | Henry Sleik, Vice-Pres., W. B. Connor Engrg., New York | D. M. Allen reported on 50th Annual Meeting in N. Y. | 45 | 0.80 |
| Manitobr | Mar. 23 | Steam | Prof. N. M. Hall, University of Manitoba | Interesting Discussion on Professor Hall's address | 38 | 1.52 |
| | Feb. 17 | Plastics | L. Rogers, Canadian General Electric Co. | F. L. Chester reported on Society's 50th Annual Meeting in New York | | |
| Massachusetts | Mar. 16 | The Flow of Air and Gas in Ducts, also General Functions of the Society | S. H. Downs, Kalamazoo, Mich., Pres. of Society | | 34 | 0.59 |

| Chapter | Meeting Date | Subject | Speakers | Other Features | Attendance | Attendance Ratio* |
|---|---|---|---|---|---|---|
| Michigan | Mar. 13 | Study of Actual vs. Predicted Cooling Load on an Air Conditioning System | J. N. Livermore, M.E., Detroit Edison Co., Detroit | W. L. Fleisher, New York, attended as guest | 54 | 0.55 |
| | | Society Goal of 5,000 Members During Next Five Years | Pres. S. S. Sanford | Chairman of Membership Committee reported on its activities | | |
| Minnesota | Apr. 3 | Activities of ASHVE | S. H. Downs, Kalamazoo, Mich., Pres. of ASHVE | Discussion on proposal to hold meeting to adopt by-laws and a constitution for Chapter. Nominating Committee reported for 1944-45 | 42 | 0.62 |
| | Mar. 13 | History of Minnesota Chapter's Discussions on Affiliation with Minnesota Federation of Engineering Societies | Prof. F. B. Rowley, Dir., Engrg. Experiment Station, University of Minnesota, and Prof. J. H. Kuhlman | Heating the Postwar Home, by Lyman Gross, Minneapolis-Honeywell Regulator Co. | 50 | 0.73 |
| Montreal | Mar. 14 | The Flow of Air and Gas in Various Systems | S. H. Downs, Pres. of ASHVE, Kalamazoo, Mich. | Thomas Worthington reported on membership | 31 | 0.50 |
| Nebraska | Mar. 14 | The History and Development of the Cooling Tower | L. T. Mart, Pres., The Marley Co., Kansas City, Mo. | Progress reports were submitted by Verne Simmonds and D. D. Williams on Program and Membership Committees | 18 | 0.95 |
| New York | Mar. 20 | Food Dehydration: Types of Systems | Prof. A. A. Giannini, Columbia University, New York | Joint Meeting with ASME Process Industries Division | 109 | 0.60 |
| | | Theory of Design | W. R. Marshall, Jr. | | | |
| | | General Application | Arnold Weisselberg | | | |
| North Texas | Mar. 20 | Omnibus Arms Case | W. H. Kuhn | W. H. Moler, Chairman of Program Committee, reported | 31 | 0.57 |
| | Feb. 28 | Unique Mechanical Features of the New Mercantile Bank Bldg. | P. N. Vinther, Consulting Engr. | J. A. Ray reported on Membership Committee. G. A. Linskie reported on the 50th Annual Meeting in New York | 42 | 0.63 |
| Northern Ohio | Mar. 13 | Air Conditioning in the Pentagon Bldg., Washington, D. C. | Charles S. Leopold, Consulting Engr., Philadelphia, Pa. | Treas. R. L. Clark submitted his report | 67 | 0.84 |
| Oklahoma | Mar. 29 | The Art of Air Recovery | Henry Sleik, Vice-Pres. W. B. Connor | Prof. E. F. Dawson, University of Oklahoma, announced night courses in refrigeration and air conditioning at Central High School | 15 | 1.37 |

| Chapter | Meeting Date | Subject | Speakers | Other Features | Attendance | Attendance Ratio* |
|---|---|---|---|---|---|---|
| Ontario | April 3 | ASHVE Activities in Wartime<br>Research | M. F. Blankin, Philadelphia, Pa.<br>Cyril Tasker, Dir. of Research, ASHVE Laboratory, Cleveland | Report of Nominating Committee was submitted | 54 | 0.38 |
| | Mar. 13 | The Flow of Air and Gas in Various Systems | S. H. Downs, Kalamazoo, Mich., Pres. of Society | A. J. Strain spoke on Membership. Nominating Committee was announced as follows: H. S. Moore, Chm., Walter Smith, C. W. Gordon | 79 | 0.54 |
| Oregon | Mar. 9 | Combustion of Fuels | Oliver W. Johnson | Officers elected as follows: Pres., E. C. Willey; Vice-Pres., F. F. Urban; Sec'y, G. H. Risley; Treas., E. A. Ponder; Board of Gov., J. A. Freeman, H. K. Mead, E. S. Turner | 30 | 0.63 |
| Pacific Northwest | Mar. 8 | Combustion of Liquid and Gas Fuels | O. W. Johnson, Standard Oil Co. of California, San Francisco | M. N. Musgrave reported on the 50th Annual Meeting in N. Y. | 42 | 0.58 |
| Philadelphia | Mar. 9 | Wartime Changes in Refrigerants | H. G. Brandt, Carrier Corp., Syracuse, N. Y. | A. C. Caldwell reported on membership. M. F. Blankin spoke on his appointment as Local Fuel Coordinator | 53 | 0.44 |
| | Feb. 10 | 1942-43 Income Tax Situation | Mr. Culp, U. S. Treasury Department | Messrs. M. F. Blankin, L. C. Davidson, W. P. Culbert and L. P. Hynes were appointed to formulate plans for formation of Board of Trustees | 43 | 0.36 |
| Pittsburgh | Mar. 13 | The Human Machine | Dr. Peter Gray, Zoologist, University of Pittsburgh | Included visit to Buhl Planetarium—"Life on Other Worlds" | 50 | 0.83 |
| South Texas | Feb. 29 | Control Equipment | D. M. Boyd, Eastern States Refining Corp., Houston, Tex. | J. A. Walsh reported on 50th Annual Meeting in N. Y. | 21 | 0.42 |
| | Jan. 26 | Heating and Ventilating Air Conditioning Code | D. S. Cooper, Chairman | Joint meeting with Houston Building Code Committee | 13 | 0.26 |
| | Dec. 29 | Dehydration of Foods and Stock Food Products | R. E. Rowe, United Gas Corp. | Committee appointed to formulate and offer assistance to City of Houston in preparation of heating, ventilating and plumbing section of proposed new building code, as follows: D. S. Cooper, Chm., B. P. Fisher, R. F. Taylor, C. A. McKinney | 18 | 0.36 |
| | Nov. 26 | Program Discussions<br>Membership and Prospects List | D. M. Mills<br>J. A. Walsh | New proposed heating code for Houston discussed | 7 | 0.14 |

| CHAPTER | MEETING DATE | SUBJECT | SPEAKERS | OTHER FEATURES | ATTENDANCE | ATTENDANCE RATIO* |
|---|---|---|---|---|---|---|
| St. Louis | April 4 | The Future of Electronics in Air Conditioning | J. E. Haines, Mgr., Air Conditioning Controls Div., Minneapolis-Honeywell Regulator Co., Minneapolis, Minn. | Report of Nominating Committee submitted<br>Movies were shown on Fundamentals of Electricity, et al. | 108 | 1.77 |
| | Mar. 7 | Radiant Heating | C. A. Hawk, Jr., A. M. Byers Co., Pittsburgh, Pa. | George Myers and M. F. Carlock were appointed to serve on Nominating Committee | 96 | 1.57 |
| Southern California | Mar. 8 | Design and Application of Strato-Chambers for Improvement of Personnel, Equipment and Material in High Altitude Flying | Maron Kennedy, Appl. Engr., York Corp., Los Angeles | Leo Hungerford gave resume of activities at 50th Annual Meeting in N. Y. | 56 | 0.75 |
| Washington, D. C. | Mar. 8 | Radiant Heating as a New Trend in Space Heating | H. K. King, A. M. Byers Co., Pittsburgh, Pa. | A. V. Hutchinson, Secretary of ASHVE, spoke briefly on Society membership | 86 | 0.50 |
| Western Michigan | Mar. 22 | Accomplishments of Past and Plans for Future of *NWAH & AC* | George Boeddener, Mgng. Dir., *NWAH & AC*, Cleveland, Ohio | T. D. Stafford gave brief outline of program for Semi-Annual Meeting of ASHVE in Grand Rapids in June | 160 | 3.33 |
| | | Past, Present and Future of Forced Warm Air Heating | Harold Mueller, Mueller Furnace Co., Pres. of *NWAH & AC* | | | |
| | | Electrostatic Precipitation of Dust (Sound Film and Lecture) | J. C. Foley, Precipitron Specialist, Westinghouse Elec. & Mfg. Co. | | | |
| | Feb. 14 | Report on papers presented at 50th Annual Meeting in N. Y. | Messrs. T. D. Stafford, L. G. Miller, C. H. Pesterfield, S. H. Downs | Prof. Miller introduced S. H. Downs, Pres.-Elect of Society | 20 | 0.42 |
| Western New York | Mar. 13 | Panel Heating<br>Axial Flow Fans<br>Electrostatic Filters | M. C. Beman<br>W. R. Heath<br>F. J. Weber | Worthwhile discussion followed each topic | 33 | 0.58 |
| | Feb. 14 | War Department Report (motion picture) | Capt. Deittre and S/Sgt. Davis | Pres. S. M. Quackenbush reported on 50th Annual Meeting in N. Y. | 26 | 0.46 |
| Wisconsin | Mar. 21 | Fuel Efficiency (coffee talk) | Parker Moe, International Harvester Co., and Regional Coordinator for the Bureau of Mines Fuel Efficiency Campaign | F. W. Goldsmith discussed affiliation with Milwaukee Engrs. Society | 60 | 1.0 |
| | | Conduit System of Air Distribution | W. S. Bodinus, Carrier Corp., Chicago | | | |
| | Feb. 28 | Practicable Methods of Accomplishing Heating Economy in Large Buildings | G. R. Bailey, Gen. Mgr., Alfred H. Wetten & Co., Chicago | Ernest Szekely reported as Chapter Delegate to 50th Annual Meeting in N. Y. | 33 | 0.55 |

*The attendance ratios shown represent the meeting attendance divided by the Chapter membership. These will be useful as a partial indication of interest shown by local chapter members in various types of subjects programmed by other local chapters and may be helpful in deciding on subjects for other chapter meetings.

## F. E. SPURNEY RETURNS TO WASHINGTON, D. C.

F. E. Spurney has recently joined the engineering staff of Luther and Wood, Washington, D. C., according to an announcement by the company. As superintendent for Alabama Drydock and Shipbuilding Co., Mobile, Ala., and Turner Construction Co. at New Orleans, Mr. Spurney has recently completed several war plants. Prior to this work he spent five years as building engineer with Federal Reserve System, Washington, D. C., and 15 years with Turner Construction Co., New York, on building construction work in the Eastern part of the country, serving as superintendent of construction and executive assistant to the vice-president.

## DEATH OF FRED BAHNSON

Word has been received of the death on March 18 of Frederic Fries Bahnson, president and consulting engineer, The Bahnson Co., and president, Southern Steel Stampings, Inc., Winston-Salem, N. C.

Mr. Bahnson was a member of the Society for over 27 years, taking an active part in Society affairs until the time of his death. He was born on March 6, 1876, at Winston-Salem, where he attended the Salem Boys Preparatory School. He entered the University of North Carolina and along with his senior year he took his first year medical, specializing in chemistry and biology. He received his Ph.B. (cum laude) in 1896. Following his graduation from the University of North Carolina, he entered the University of Pennsylvania Medical School, but after six weeks of study was forced to give it up on account of eye strain. He rested a year and then entered the electrical field.

His shop experience was obtained with the Stanley Electric Mfg. Co., Pittsfield, Mass., where he became testing and erecting engineer, and was associated with the development work of Alexander Graham Bell and Marconi. In 1903 he was made chief engineer for the Florida East Coast Hotel Co. at Nassau, Bahamas, and in 1904 he was chief electrician for them at Miami and Palm Beach, Fla. He then spent a year with John W. Fries, Winston-Salem, N. C., on development and research work on air conditioning, and in 1906 became manager of the F. and H. Fries Wool Mill at Winston-Salem, where he remained for six years. During these years he was closely associated with John W. Fries on additional research in air conditioning and on wool drying in mills.

He spent two years with Briggs-Sheffner Co., Winston-Salem, handling humidifiers under John W. Fries Patents. He also acted as engineer, salesman and as treasurer for six months. In 1914 he became engineer for Normalair Co., engaged in the manufacture and sale of humidifiers and air conditioning apparatus with automatic control of temperature and humidity. He had sole charge of designing and erecting all installations, specializing in research and development along with design and erection of plants.

In 1919 Mr. Bahnson became vice-president and engineer for the Bahnson Humidifier Co., later known as The Bahnson Co. In 1932 he was made president of the Southern Steel Stampings, Inc. In 1940 he became consulting engineer for The Bahnson Co., maintaining both these positions up until his death.

Mr. Bahnson designed and placed on the market a conditioning room which was very successful depending on air circulation, heat and humidity.

In 1918-19 Mr. Bahnson served as a member of the ASHVE Committee to Confer with the Weather Bureau. He contributed several papers which were presented before Society Meetings.

Mr. Bahnson was known not only as a genius in engineering, but as a leader in fraternal circles. He was a member of Winston Lodge 167, A. F. and A. M., Winston Chapter 24, Royal Arch Masons, Piedmont Commandery 6, Knights Templar, the Greensboro Consistory of Scottish Rite Masons, and the Grand Council of Knight Masons of Dublin, and at the time of his death was Provincial Grand Superintendent.

Active in civic affairs, he was a charter member of the Kiwanis Club, a member of the Twin City and Forsyth Country Clubs, the Y.M.C.A., and was a life member of the *American Society of Mechanical Engineers*. He was the only American member of Great Chief's Council of Knight Masons of Dublin, Ireland, and was among those listed in "Who's Who in Engineering."

Mr. Bahnson was an active member of Home Moravian Church, having served on numerous boards of the church and took a great interest in the Men's Bible Class, as assistant to the teacher, Dr. Howard Rondthaler.

Mr. Bahnson was married in Charlotte on April 14, 1910, to Miss Bleeker Reid, who survives. The Officers and Council of the Society extend their sincere and heartfelt sympathy to Mrs. Bahnson, and to her four sons, Frederic F. Bahnson, Jr., Winston-Salem, Reid of the Medical Corps, USNR, at present in the South Pacific, Henry of the Naval Reserve V-12 program at Harvard Medical School, and Alex, a member of the mountain troops stationed at Camp Hale, Col.; also to his mother, and two sisters, Mrs. T. Holt Haywood and Mrs. James A. Gray, and a brother, A. H. Bahnson, all of whom are from Winston-Salem.

Funeral services were conducted at Home Moravian Church on Sunday, March 19, with burial in the Moravian Graveyard, Winston-Salem.

## J. W. BASSETT DIES IN DETROIT

The Michigan Chapter of the Society announces with regret the death of James Wylie Bassett on April 7, 1944, at the age of 57. Mr. Bassett was born in Birmingham, Mich., April 7, 1887. He attended the Township Public School and the Birmingham High School.

His first position was shipping clerk for the Walker Mfg. and Supply Co., Detroit, and after a short time he became salesman of plumbing and heating supplies, remaining with them for ten years. In 1917 he was employed as estimator and salesman by the McCormick Plumbing Supply Co., Detroit, which he maintained until 1920, when he joined the staff of W. A. Case and Sons, also in Detroit.

After five years he became associated with Fred Barton, acting as estimator and salesman, which position he held for four years. In 1930 he became sales engineer in charge of the Detroit office of McQuay, Inc., remaining at this post until his death.

Mr. Bassett joined the Society in 1938 and had been actively engaged in the heating and ventilating industry for many years in Detroit and the surrounding districts.

Among his associates in the Michigan Chapter he was considered an excellent sales engineer, with a reputation for honesty and fair play, which brought him a host of friends and the universal respect of his fellow members.

The Officers and Council of the Society extend their deepest sympathy to his widow, Mrs. Mabel Bassett, his son, James Camp Bassett, and to his brother and three sisters.

## PROF. R. A. NORMAN, IOWA STATE COLLEGE DIES

Prof. Roy A. Norman, 715 Ridgewood Ave., Ames, Ia., died January 29, 1944, at the Mary Greeley hospital, after an illness of three months.

Professor Norman was born February 13, 1877, in Logan, Ia., where he received his early education. He was graduated from the Logan High School and then attended Iowa State College, from which he received his B.S. in 1903 and his M.E. degree in 1909.

From 1903 to 1907 he was employed by the Milwaukee Railroad and then for a short time he worked for the American Locomotive Co. In the fall of 1907 he was appointed Assistant Professor at Iowa State College, which title he retained until 1917. From 1919 until 1925 he was an Associate Professor in Mechanical Engineering at Iowa State and in 1925 he was appointed Professor, and remained at the College until his death.

Professor Norman became a member of the Society in 1937, and was very active in organizing the Iowa-Nebraska Chapter of the Society. When the Iowa Chapter was organized, Professor Norman was elected its first president.

Besides being a member of the ASHVE, he was a member of the *ASME*, Iowa Engineering Society, Kiwanis Club, Tau Beta Pi and Pi Tau Sigma, honorary engineering fraternities. He was also active in Masonic work, being a member of the A.F. and A.M. No. 249, Royal Arch Masons, Knights Templar, and a member of the council. Professor Norman was active in church work, and was a life deacon in the Ames Baptist Church.

The Officers and Council of the Society extend their sincerest sympathy to his family who survive.

# Candidates for Membership

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 49 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by May 15, 1944, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

| CANDIDATES | REFERENCES | |
|---|---|---|
| | *Proposers* | *Seconders* |
| ALYEA, HAROLD W., Design & Development Engr., Johnson Service Co., Milwaukee, Wis. | J. R. Vernon<br>J. A. Cutler | H. W. Schreiber<br>O. A. Trostel |
| AVERA, ALVAN F., Chief Engr., Dallas Air Conditioning Co., Dallas, Tex. | M. L. Brown<br>L. C. McClanahan | J. A. Ray<br>E. J. Stern |
| BARTHELMESS, RICHARD, Mech. Engr., War Department, U. S. Engineers, Jacksonville, Fla. | J. C. Pastor<br>F. J. Beckwith | C. W. Stewart<br>E. N. Sanbern |
| BEAN, THOMAS H., JR., Utilization Engr., New Orleans Public Service, Inc., New Orleans, La. | G. E. May<br>J. S. Burke | L. V. Cressy<br>W. H. Grant, Jr. |
| BELANGER, WILLIAM, Owner, Southeastern Supply Co., New Bedford, Mass. | D. M. Archer<br>C. T. Flint | C. W. Kimball<br>C. W. Larson |
| BOUCHER, KENT D., Mgr., Carrier Div., United Clay Products Co., Washington, D. C. | W. C. Jones<br>W. H. Littleford | L. Maurer<br>S. L. Gregg |
| BOYD, LYLE E., Air Conditioning Supervisor, Caterpillar Military Engine Co., Decatur, Ill. *(Advancement)* | W. L. Cassell<br>P. C. Leffel | W. E. Howarth<br>D. M. Allen |
| CRONSTROM, KENNETH A., Owner, Cronstrom Furnace & Sheet Metal Works, Minneapolis, Minn. | C. P. Petersen<br>C. T. Lawrence | H. M. Betts<br>J. F. Stafford |
| DAVIES, EDWIN A., Dist. Repr., Hoffman Specialty Co., Des Moines, Ia. | F. Jehle<br>F. H. Gaylord | C. W. Stewart<br>E. N. Sanbern |
| DUBE, JOHN E., Vice-Pres. & Dir., Alco Valve Co., St. Louis, Mo. | C. F. Boester<br>B. C. Simons | G. E. Roepke<br>*F. M. MacDougall |
| FARRELL, GEORGE F., Htg. & Vtg. Engr., Lake City Ordnance Plant, Independence, Mo. | S. Furber<br>D. M. Allen | C. Clegg<br>P. C. Leffel |

GARDNER, FRANCIS B., Secy-Treas., H. S. McClelland, Inc., Los Angeles, Calif.

HARMONAY, WILLIAM L., Pres., Michael Harmonay Corp., Yonkers, N. Y. *(Advancement)*

HAUF, JOSEPH C., JR., Engr., May Oil Burner Corp., Baltimore, Md.

HECKERT, WILLARD H., Htg. & Fuel Oil Div. Mgr., American Lubricants, Inc., Buffalo, N. Y.

HILMER, RALPH H., Engr., C. G. Hokanson Co., Los Angeles, Calif.

KALLUSCH, HOWARD A., Precipitron Appl. Engr., Westinghouse Electric & Manufacturing Co., Cleveland, Ohio.

KAVANAGH, PERRY E., Administrative Asst., Yale University, New Haven, Conn.

KUHN, WILLIAM H., Partner, Payne & Kuhn, Dallas, Tex.

LEDBETTER, NORVELL C., Htg. Engr., N. O. Nelson Co., Memphis, Tenn.

LEITCH, KELVIN D., Treas., The Arthur S. Leitch Co., Ltd., Toronto, Ont., Canada.

LEWIS, J. C., Factory Repr., Drayer & Hanson, Inc., and Farr Co., Bellaire, Tex.

L'HOMMEDIEU, CURTISS L., Htg. & Vtg. Engr., The Foskett & Bishop Co., New Haven, Conn.

LUSH, CLIFFORD N., Mech. Supt., Weston's Bread & Cake (Canada), Ltd., Winnipeg, Man., Canada.

MCDONALD, FRANK L., Branch Mgr., Barber-Colman Co., Los Angeles, Calif.

MILLER, NORTON R., Sales & Application Engr., The Trane Co., Hollis, L. I., N. Y.

MOE, PARKER A., Chief Engr., International Harvester Co., Milwaukee, Wis.

MUSGRAVE, MERRILL N., Owner, Merrill N. Musgrave & Co., Seattle, Wash. *(Advancement)*

NELSON, NORRIS T., Vtg. Engr., Bethlehem Steel Co., San Francisco, Calif.

OLSON, ARTHUR A., Vice-Pres., Lee Engineering Co., Youngstown, Ohio.

OOSTEN, LOUIS S., Asst. Chief Engr., Bell & Gossett Co., Morton Grove, Ill. *(Advancement)*

ORR, WELDON J., Draftsman, Jas. A. Kearns, Montreal, Que., Canada.

OTIS, ALFRED L., Dist. Sales Mgr., Clarage Fan Co., West Hartford, Conn.

RUNGE, EDWIN A., Chief Engr., Lee Engineering Co., Youngstown, Ohio.

SAVAGE, SIDNEY W., Engr., Gilson Manufacturing Co., Ltd., Toronto, Ont., Canada.

SHEA, FRANCIS X., Eastern Repr., Farr Co., Rego Park, L. I., N. Y.

SHEARER, ARTHUR G., Mgr., Insulation Div., Hou-Tex Roofing Co., Houston, Tex.

SMERLING, DANIEL, Dist. Sales Engr., Flynn & Emrich Co., New Haven, Conn.

SMITH, CHARLES H., Secy., L. J. Wing Manufacturing Co., New York, N. Y.

SMITH, HOWARD W., JR., Engr., Southern California Telephone Co., Los Angeles, Calif.

SMITH, NORMAN A., Pres., Norman A. Smith Co., Toronto, Ont., Canada.

SPARKS, JAMES D., Northwest Mgr., Ilg Electric Ventilating Co., Seattle, Wash. *(Advancement)*

TOLERTON, HILL A., Contracting Engr., Lee Engineering Co., Youngstown, Ohio.

WAGENBRENNER, LOUIS H., Chief Mech. Engr., Ralph E. Phillips, Los Angeles, Calif.

WELLFORD, WALTER L., JR., Secy., Chickasaw Wood Products Co., Memphis, Tenn.

WHYBREW, GEORGE H., Chief Mech. Engr., Giffels & Vallet, Inc., Detroit, Mich.

WIELAND, JOHN J., Mech. Engr., Alto, Mich.

WOODSON, EDWARD G., Mgr. Memphis Office, The Trane Co., Memphis, Tenn.

ZWERLING, SEYMOUR J., JR., Engr., Triangle Ventilating Co., Brooklyn, N. Y.

| | |
|---|---|
| W. O. Stewart | Art Theobald |
| Leo Hungerford | R. A. Lowe |
| W. F. Rainger | A. A. Abrams |
| G. E. Olsen | A. R. Herske |
| J. F. Collins, Jr. | L. G. Vance |
| L. W. McCrea | James Posey |
| B. C. Candee | J. Davis |
| J. A. Collins | S. M. Quackenbush |
| C. G. Hokanson | Art Theobald |
| Leo Hungerford | R. A. Lowe |
| J. C. Fitz | J. L. Foley |
| I. R. Cummings | *W. F. Begoon |
| L. A. Teasdale | L. E. Seeley |
| W. W. Rundo* | T. J. Converse |
| H. J. Martyn | J. A. Ray |
| E. T. Gessell | C. R. Gardner |
| J. D. Flinn | J. J. Nolan |
| R. H. Hoshall | W. A. Danielson |
| A. S. Leitch | V. J. Jenkinson |
| W. J. Usher, Jr. | A. J. Strain |
| J. A. Walsh | W. J. Way |
| D. M. Mills | L. D. Greenwood |
| T. J. Converse | J. R. Smak |
| W. Roeder | W. K. Simpson |
| P. L. Charles | R. L. Kent |
| E. Anderson | D. F. Michie |
| W. O. Stewart | R. A. Lowe |
| Leo Hungerford | A. Theobald |
| A. Baum | D. G. Milligan |
| V. J. Cucci | E. Sternberg |
| Rex Vernon | T. M. Hughey |
| F. W. Goldsmith | H. W. Schreiber |
| J. D. Sparks | M. J. Hauan |
| R. D. Morse | S. D. Peterson |
| A. L. Walters | F. E. Triggs |
| W. J. Hennessy | T. R. Johnson |
| L. E. Slawson | R. A. Wilson |
| W. M. Rowe | D. L. Taze |
| E. J. Gossett | C. E. Pullum |
| R. E. Moore | H. A. Lockhart |
| F. Phipps | Leo Garneau |
| *C. W. Lachopelle | Aylmer Hamlet |
| S. H. Downs | J. R. Smak |
| W. Roeder | G. R. Couch |
| L. E. Slawson | R. A. Wilson |
| W. M. Rowe | D. L. Taze |
| F. W. Stott | V. J. Jenkinson |
| A. Karges | W. H. Evans |
| J. W. Markert | J. H. Fogg |
| V. J. Cucci | A. E. DeSomma |
| A. B. Banowsky | F. C. Brandt |
| C. A. McKinney | A. F. Barnes |
| T. J. Converse | J. R. Smak |
| L. A. Teasdale | W. Roeder |
| R. A. Wasson | Harry Schmidt |
| H. S. Wheller | J. H. Carbone |
| Leo Hungerford | Art Theobald |
| W. O. Stewart | R. A. Lowe |
| F. W. Chambers | R. W. Bayles |
| Robert Kerr | A. J. Strain |
| S. D. Peterson | R. D. Morse |
| M. N. Musgrave | M. J. Hauan |
| L. E. Slawson | L. T. Avery |
| W. M. Rowe | R. A. Wilson |
| W. O. Stewart | Leo Hungerford |
| R. E. Phillips | Art Theobald |
| J. D. Flinn | W. A. Danielson |
| W. E. Thorpe | J. J. Nolan |
| *E. R. Hewett | *T. Taylor |
| E. F. Hyde | R. S. M. Wilde |
| W. W. Bradfield | F. C. Warren |
| T. D. Stafford | *P. O. Wierenga |
| J. J. Nolan | R. H. Hoshall |
| W. A. Danielson | W. E. Thorpe |
| M. C. Giannini | *A. C. Coonradt |
| *W. A. Rose | *L. Sternberg |

*Non-Member.

# Candidates Elected

In the past issues of the Journal of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. III, Sec. 8, of the By-Laws, the following list of candidates elected:

## MEMBERS

ACHESON, LOUIS K., Head, Mechanical Research Laboratory, The Hoover Co., North Canton, Ohio.

ALEXANDER, GRANVILLE P., Secy., Heating, Piping & Air Conditioning Contractors Philadelphia Association, Philadelphia, Pa.

AMMERMAN, ANDREW S., JR., Dist. Mgr., Aerofin Corp., Chicago, Ill. *(Advancement)*

ANDERSON, CARL O., Htg. & Vtg. Engr., Board of Education, Chicago, Ill.

BEST, E. THOMPSON, Mgr., Air Cond. & Refrig., Frank A. McBride Co., Paterson, N. J.

BOELTER, L. M. K., Assoc. Dean, College of Engineering, University of California, Berkeley, Calif.

BRADFIELD, JOSEPH L., Works Mgr., Best Universal Lock Co., Inc., Indianapolis, Ind.

BRIGHAM, FREDERICK H., Engr., C. R. Swaney Co., Boston, Mass. *(Reinstatement)*

CARTIER, MARCEL E., Research Engr., American Radiator & Standard Sanitary Corp., Yonkers, N. Y.

CHAMBERS, S. A., Vice-Pres., Almirall & Co., Inc., New York, N. Y.

CLARK, L. O. RAY, Dist. Mgr., B. F. Sturtevant Co., Hartford, Conn.

CONNELL, E. C., Htg. Engr., Sullivan Valve & Engineering Co., Butte, Mont.

CUMMINGS, GEORGE C., New York Repr., Dravo Corp., New York, N. Y.

DARBY, F. NORMAN, Engr., Sun Shipbuilding & Drydock Co., Chester, Pa.

DARRAH, RICHARD D., Sales Engr., Dravo Corp., Pittsburgh, Pa.

DOERRFUSS, HARRY W., Sales Engr., Warren Webster Co., Philadelphia, Pa.

DONCEEL, G. T., Indus. Engr., Oklahoma Natural Gas Co., Oklahoma City, Okla.

ELDRED, HAROLD, Htg. & Vtg. Contractor, North Wembley, England.

GAGE, BURFORD B., Owner, Gage Plumbing & Heating Co., Salina, Kans.

HILL, N. E., Designing Engr., The L. J. Mueller Furnace Co., Milwaukee, Wis.

HOWARTH, WALTON E., Designing Engr., William L. Cassell, Kansas City, Mo.

HULL, E. O., Engr. Designer, Skidmore, Owings, Merrill & Andrews, Oak Ridge, Tenn.

HUNT, JAMES F., Mech. Engr., Cory & Joslin, Inc., San Francisco, Calif.

JANSSEN, HENRY J., Treas., Child & Scott-Donohue, Inc., New York, N. Y.

KELL, JOHN R., Chief Htg. & Vtg. Engr., Dr. Oscar Faber, St. Albans, Herts, England.

KOTZEBUE, R. W., Head, Equipment Control Unit, Fire Prevention Section, Headquarters Eighth Service Command, Dallas, Tex. *(Reinstatement and Advancement)*

LUBKING, CHARLES J., Engr., Minneapolis-Honeywell Regulator Co., Philadelphia, Pa.

LUZZI, THEODORE E., Engr., Almirall & Co., Inc., Falls Church, Va.

MASON, ROBERT E., Owner, Robert E. Mason & Co., Charlotte, N. C.

MCGEARY, PAUL W., Mgr., Indiana District Office, The Trane Co., Indianapolis, Ind.

MEDCALF, LLOYD C., Consulting Engr., Mt. Vernon, N. Y.

MUTH, ARNOLD J., Elec. Engr., Albert Kahn Associated Architects & Engineers, Inc., Detroit, Mich.

NATKIN, A. J., Partner, Natkin & Co., Houston, Tex.

NELSON, ARTHUR W., Major C. E., Post Engr., Camp Edwards, Mass. *(Reinstatement and Advancement)*

NEVIN, JOHN F., Capt., % Chief Engineer, Eastern Command, H.Q., India.

NOTKIN, JAMES B., Mech. Engr., Puget Sound Navy Yard, Bremerton, Wash.

OLDS, A. DEAN, Chief Engr., Coleman Lamp & Stove Co., Wichita, Kans.

PARKES, C. H., Distributor of Heating & Pumping Equipment, Dodge City, Kans.

PAULDING, LEWIS G., Supervisor, Long Island Lighting Co., Mineola, L. I., N. Y. *(Reinstatement)*

PETERS, WILLIAM S., Partner & Gen. Mgr., A. J. Peters & Son, San Jose, Calif.

PHILLIPS, MAURICE E., Mgr., Air Conditioning & Htg. Div., United Electric Service, Monroe, La.

PIEKSEN, GEORGE W., Sales Engr. and Repr., Sarco Co., Inc., and Spence Engineering Co., St. Louis, Mo.

REED, THOMAS M., JR., Sales Engr. for John J. Nesbitt, Inc., New York, N. Y.

RITTER, IRVING S., Design Engr., Westinghouse Electric Elevator Co., Jersey City, N. J.

ROLAND, JOHN, Chief Engr., Htg. Div., Schwitzer-Cummins Co., Indianapolis, Ind.

RUSSELL, WILLIAM B., Asst. to Pres., Kewanee Boiler Corp., Kewanee, Ill. *(Reinstatement)*

SAUNDERS, SIDNEY G., Civilian Staff Officer (Mechanical Engineering), Banstead, Surrey, England.

SCHULMEISTER, WALTER A., Sales Mgr., American District Steam Co., North Tonawanda, N. Y.

SLUDER, CLARENCE T., Mech. Engr., Federal Housing Administration, Washington, D. C.

STEIN, HERMAN W., Dir. of Research, Engrg. Dept., Celotex Corp., Chicago, Ill.

STELL, WILLIAM H., JR., Owner, Arnold R. Kamman Co., Buffalo, N. Y.

STEVENS, JOSEPH W., Dist. Mgr., Minneapolis-Honeywell Regulator Co., Syracuse, N. Y.

TAGGART, ROBERT F., Mech. Engr., O. W. Motz, Cincinnati, Ohio.

TREFTS, JOHN C., JR., Vice-Pres., Farrar & Trefts, Inc., Buffalo, N. Y.

URMSTON, BENJAMIN S., Dist. Engr., Airtemp Division of Chrysler Corp., Dallas, Tex.

VERITY, ELBERT W., Pres., Almirall & Co., Inc., New York, N. Y.

WEAVER, EUGENE A., Sales Mgr., Space Heating Division, Surface Combustion, Toledo, Ohio.

WHEELER, HAROLD E., Pres., Air Comfort Corp., Chicago, Ill.

WHITTINGTON, WILLIAM P., Prop., Whittington Pump & Engineering Co., Indianapolis, Ind. *(Reinstatement)*

WILSON, ARTHUR A., Capt., U.S.M.C.R., Camp Refrig. Engr., U. S. Marine Corps, Camp Lejeune, New River, N. C.

ZAVORSKI, LEONARD, Vice-Pres. in charge of Production, Tuttle & Bailey, Inc., New Britain, Conn.

## ASSOCIATES

ADAMS, A. K., Mgr., Montana Iron Works, Butte, Mont.

ALMERT, HAROLD, Consulting Engr., New York, N. Y.

BAAK, WALTER J., Branch Mgr., Minneapolis-Honeywell Regulator Co., Omaha, Nebr.

BAIRD, A. D. G., Pres., A. G. Baird, Ltd., Toronto, Ont., Canada.

BARAGAR, J. S., Head of Office, DeWitt C. Griffin & Associates, Spokane, Wash.

BARNETT, ROBERT E., Engr., L. D. Shipbuilding Co., Sturgeon Bay, Wis.

BELYEA, ROY E., Mgr., Belyea Bros., Ltd., Toronto, Ont., Canada.

BUSCHMANN, ALFRED W., Mgr., Buschmann Co., Indianapolis, Ind.

CANNON, VERNON K., Capt., Asst. Chief, Supply Training Branch, Ft. Francis E. Warren, Wyo.

CHANDLER, ROY, Export Mgr., William H. Wheeler & Co., New York, N. Y.

CLARK, H. C., Mfrs.' Agent, Houston, Tex.

COOPER, C. H., Htg. Asst., Chase Co., Eugene, Ore.

COOPER, GEORGE P., Combustion Engr., Empire-Hanna Coal Co., Ltd., Toronto, Ont., Canada.

COUCH, GEORGE R., Owner, George R. Couch Co., Hartford, Conn.

DARLING, ROBERT B., Pres., Robert B. Darling Co., Inc., New York, N. Y.

DUNN, ROBERT, Chief Engr., The Canadian Bank of Commerce, Toronto, Ont., Canada.

GRISWELL, HENRY D., Vice-Pres., Hosmer Stokol Corp., Indianapolis, Ind.

HAGUE, WILLIAM S., Mgr., Plbg., Htg. & Indus. Supplies, Crane Co., Indianapolis, Ind.

HARDER, E. PAUL, Htg. & Vtg. Engr., Emerson Electric Manufacturing Co., St. Louis, Mo.

HARRINGTON, JEREMIAH P., Mech. Engr., U. S. Engineers, Omaha, Nebr.

HEATH, GEORGE A., Sales Engr., Mechanicsburg, Pa.

HOLLIS, RICHARD C., Sales Engr., Consolidated Conditioning Corp., Mt. Vernon, N. Y.

HOSMER, GEORGE H., Pres., The Hosmer Stokol Corp., Indianapolis, Ind.

HOSTETTER, JOHN C., Pres., Mississippi Glass Co., St. Louis, Mo.

JACK, ROBERT A., Keeney Publishing Co., Cleveland Heights, Ohio.

LAMBERT, EVERETT C., Air Conditioning & Ind. Refrig. Supvr., Westinghouse Electric Elevator Co., Philadelphia, Pa.

LANGE, CLIFFORD C., Htg. Engrg. Supvr., Sullivan Valve & Engineering Co., Butte, Mont.

LESTER, E. ARCHER, Asst. to Mgr., Crane Co., Indianapolis, Ind.

MANN, LEE B., Vice-Pres., Boston Filter Co., Inc., Charlestown, Mass. *(Reinstatement)*

MARSDEN, EDWARD C., Pres., Marsden & Wasserman, Inc., Hartford, Conn.

MCMAHON, LOUIS A., Htg. Engr., American Furnace Co., St. Louis, Mo.

MAXWELL, THOMAS N., Mgr., Northwest Supply Co., Butte, Mont.

MEATES, RICHARD F., Chief Refrig. Mechanic, U. S. Army, San Francisco, Calif.

METCALF, RALPH H., Project Engr., Minneapolis-Honeywell Regulator Co., Minneapolis, Minn. *(Reinstatement)*

MORROW, ALBERT P., Engr., Haynes Blankin Corp., Philadelphia, Pa.

MUELLER, E. J., Sales Engr., Sullivan Valve & Engineering Co., Butte, Mont.

MURRAY, WILLIAM W., JR., Vice-Pres., Almirall & Co., Inc., New York, N. Y.

O'DONNELL, LAWRENCE D., Partner & Mgr., McCarthy's Sheet Metal Works, Vincennes, Ind.

PALMER, ELBERT M., Mgr., Government Dept., Kewanee Boiler Corp., Kewanee, Ill.

PLATTS, EDMUND M., Vice-Pres., La-Del Conveyor & Manufacturing Co., New Philadelphia, Ohio.

PYSHER, MAURICE W., Sales Mgr., Standard Supply Co., Portland, Ore.

ROEPKE, GEORGE E., Engr. in charge of Refrig., Alco Valve Co., Kirkwood, Mo.

SCOTT, ANGUS C., Engrg. Dept. Repr., Hall-Neal Furnace Co., Indianapolis, Ind.

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